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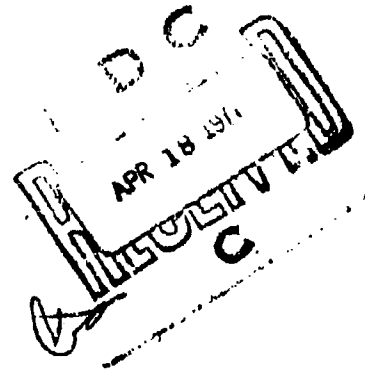
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# FLUIDIC GUN STABILIZATION SYSTEM DEVELOPMENT PROGRAM

CR-77-081-1—Fluidic Gun Stabilization System Development Program,  
by Thomas B. Tippetts and Wallis T. Fleming

MARCH 1977



Prepared by

AIRESEARCH MANUFACTURING COMPANY  
A DIVISION OF THE GARRETT CORPORATION  
402 SOUTH 36TH STREET  
PHOENIX, ARIZONA 85034

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)  This report describes the results of a design and development program conducted by AiResearch Manufacturing Company which was intended to demonstrate the feasibility of a fluidic gun stabilization system for the elevation axis. The system utilizes an angular accelerometer with pneumatic pickoff, a pneumatic/fluidic dynamic compensation and amplification circuit, and a pneumatic/hydraulic pressure control servovalve driving the hydraulic gun			

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## 20. ABSTRACT (CONCLUDED)

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actuator. The servovalve, handle valve, actuator, turret test fixture, and vehicle data used in design of the system were related to the MIC-V tank and were provided by Rock Island Arsenal.

The tasks accomplished during the program included the following:

- o Development of system analytical models;
  - o Design and fabrication of a breadboard accelerometer-controller unit;
  - o Testing of controller components; and
  - o Demonstration of closed loop system performance for the elevation axis while installed in a full-scale turret test fixture.
- 

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## FOREWORD

This is the final report by AiResearch Manufacturing Company of Arizona, a Division of The Garrett Corporation, under HDL Contract DAAG39-76-C-0081. The program reported herein was concerned with the development and feasibility demonstration of a fluidic gun stabilization system. The system, identified as AiResearch System 710538-1-1, is intended for installation in combat vehicles such as tanks and armored personnel carriers to assist the gunner in maintaining the gun properly aligned with the selected target while the vehicle is in motion over uneven terrain and the gun mounting is being subjected to random disturbances through the vehicle suspension system.

## 1. INTRODUCTION

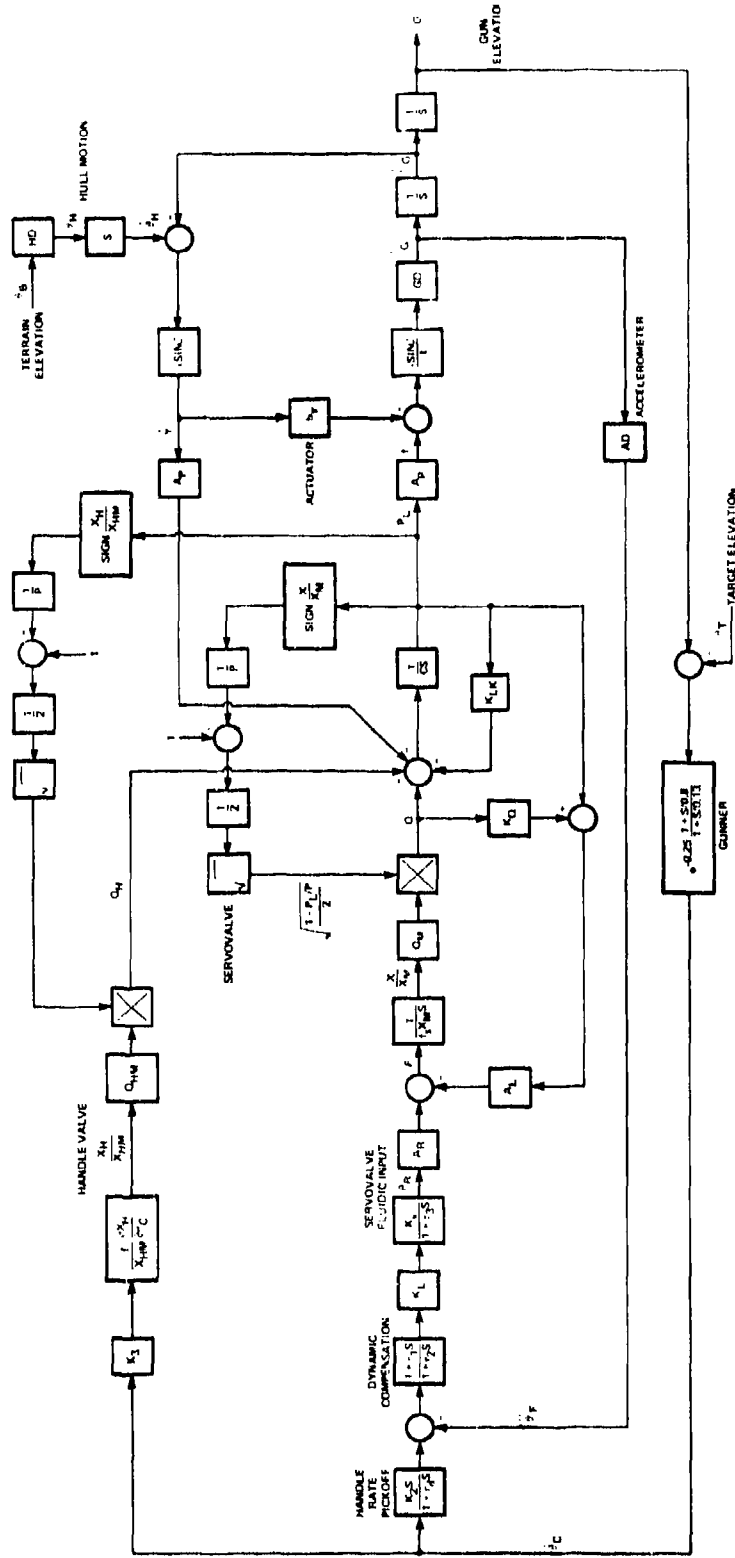
The fluidic gun stabilization system described in this report was designed for the elevation axis only. However, the system concepts and control techniques developed for the elevation axis are also applicable to the azimuth axis, thus providing the technology for development of a completely fluidic gun control system. AiResearch has previously developed breadboard components for an azimuth axis control for the M60A1 tank under Rock Island Arsenal Contract DAAA09-74-C-2069. The elevation axis control concept employed for analytical and test evaluation consisted of an angular accelerometer and fluidic controller to attenuate external disturbances and to improve control of the elevation alignment of the gun.

## 2. SYSTEM ANALYSIS

The following aspects of the fluidic gun stabilization system were explored analytically:

- A linear frequency response analysis was performed on the accelerometer-controller-gun actuation loop to gain insight into the general dynamic performance of the system.
- A linear frequency response analysis was performed to determine the effect of controller noise on tracking accuracy.
- An analog computer model of the system, which included system nonlinearities and gunner dynamic performance, was studied to obtain information on overall system performance.

Figure 1 presents a block diagram of the tank gun elevation control system which utilizes two hydraulic valves. Inputs from the gunner enter the system via the control handle which is connected directly to the gunner's control valve. The tracking velocity of the gun is controlled by displacement of the handle. The second valve is a pneumatic/hydraulic pressure control servovalve, driven by the dynamically compensated output of the fluidic accelerometer. Also included is a handle rate pickoff which functions to suppress the accelerometer feedback signal when the gunner desires to change the commanded tracking rate. The dynamic transfer function representing the gunner was obtained by averaging the time constants contained in the first two functions on Page 350 of Reference 1. Because of the



PARAMETERS			VARIABLES		
SYMBOLS	UNITS	DESCRIPTION	SYMBOLS	UNITS	DESCRIPTION
$A_h$	6.862 IN. <sup>2</sup>	VALVE SPOOL COMMAND LAND AREA	$T$	RAD	TARGET ELEVATION ANGLE
$A_L$	0.0005 IN. <sup>2</sup>	VALVE SPOOL FEEDBACK LAND AREA	$x_g$	RAD	GUN ELEVATION ANGLE
$f_s$	LB/SEC	VALVE SPOOL FRICTION	$x_h$	RAD	TERRAIN ELEVATION ANGLE
$x_h$	IN.	MAXIMUM SPOOL STROKE (SERVOVALVE)	$\dot{x}_h$	RAD/SEC	HULL ELEVATION RATE
$\dot{x}_h$	IN./SEC	MAXIMUM VALVE FLOW RATE (SERVOVALVE)	$\ddot{x}_h$	RAD/SEC <sup>2</sup>	INDICATED GUN ACCELERATION
$K_D$	20 POUNDS	SERVOVALVE DROOP CONSTANT	$\ddot{x}_g$	RAD/SEC <sup>2</sup>	COMMAND GUN ELEVATION RATE
$P$	250 PSI	HYDRAULIC SUPPLY PRESSURE	$\ddot{x}_g$	PSID	COMMAND ACTUATOR PRESSURE
$C$	$3.25 \times 10^{-8}$ IN. <sup>2</sup> /LB	HYDRAULIC CAPACITANCE	$P_L$	PSID	ACTUATOR PRESSURE
$A_p$	2.0 IN. <sup>2</sup>	ACTUATOR PISTON AREA	$F$	LB	SPOOL FORCE
$I$	$39.7$ LB-FT/SEC <sup>2</sup>	GUN MOMENT OF INERTIA	$x$	IN.	SPOOL DISPLACEMENT
$\eta_p$	0.4 IN. LB/IN. SEC	FRICTION REFERRED TO GUN ANGLE	$\dot{x}$	IN./SEC	HYDRAULIC FLOW RATE
$\eta_v$	0.15 LB/SEC/IN.	FRICTION REFERRED TO ACTUATOR DISPLACEMENT	$y$	IN./SEC	ACTUATOR VELOCITY
$C_{GM}$	0.5 IN.	ACTUATOR MOMENT ARM	$\dot{Q}_h$	IN./SEC	HANDLE VALVE FLOW
$G_D$	DIMENSIONLESS	GUN BENDING DYNAMICS	$x_h$	IN.	HANDLE VALVE STROKE
$H_D$	DIMENSIONLESS	HULL SUSPENSION DYNAMICS	$s$	1/SEC	LAPLACE VARIABLE
$A_D$	DIMENSIONLESS	ACCELEROMETER DYNAMICS			

FIGURE 1  
HDL/RIA GUN STABILIZATION SYSTEM  
MIC-V TANK WITH 20MM GUN  
MOOG SERVOVALVE MODEL 15-010  
4-16-76 (REV. A 4-23-76)

presumed ability of the gunner to adapt himself to a particular control task, the gain in the gunner transfer function was adjusted to a value that gave optimum response and stability.

To concentrate the analysis on those parameters which have a significant effect on system performance, the system block diagram was reduced to that shown on Figure 2. The procedure and justification for this reduction is outlined in Appendix A.

## 2.1 Linear System Analysis

Dynamic performance of the gun stabilization system was explored by means of a frequency response analysis of the accelerometer loop shown in Figure 2. The effects on disturbance suppression of variations in the lead time constant,  $\tau_1$ , the lag time constant,  $\tau_2$ , and the servovalve time constant,  $\tau_3$ , were evaluated.

Disturbance suppression is given by the following function (for  $\theta_C = 0$ ):

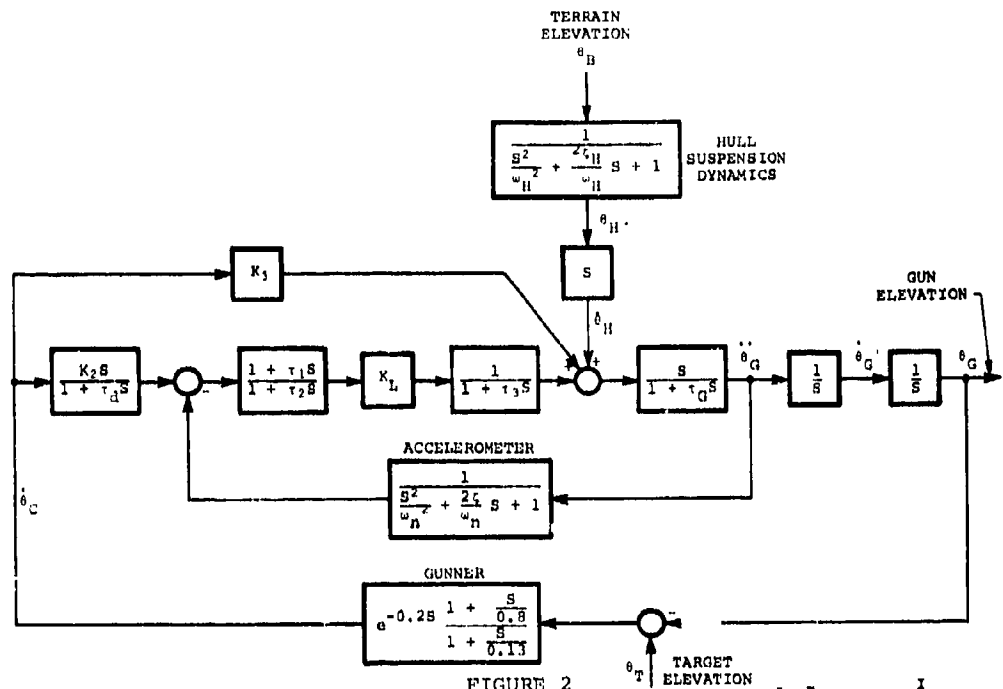
$$\frac{\theta_G}{\theta_H} = \frac{1}{(1 + \tau_G S)(1 + K_L GH)}$$

where

$$GH = \frac{(1 + \tau_1 S)(S)}{(1 + \tau_G S)(1 + \tau_2 S)(1 + \tau_3 S) \left( \frac{S^2}{\omega_n^2} + 2\zeta \frac{S}{\omega_n} + 1 \right)}$$

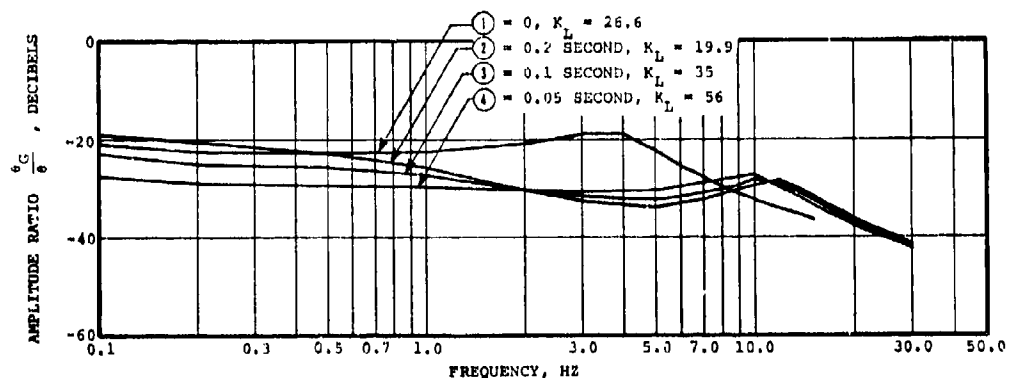
The amplitude of this function is plotted versus frequency in Figures 3, 4, and 5 for various values of  $\tau_1$ ,  $\tau_2$ , and  $\tau_3$ , respectively. The loop gain,  $K_L$ , was set at a value which gave a stability gain margin of 10 db for each case. The effectiveness of the stabilization system in suppressing disturbances is measured by the amount that the curves fall below zero decibels.

Figure 3 illustrates the effect of the lead time constant on disturbance suppression. Comparison of Curve 1 with Curve 4 on Figure 3 demonstrates that omission of the lead factor results in significant degradation. In other calculations, no significant improvement was noted when other parameters were varied as long as the value of  $\tau_1$  remained equal to zero. Inclusion of a lead time constant of  $\tau_1 = 0.05$  second resulted in the optimum suppression which is depicted in Curve 4 on Figure 3.



LINEARIZED BLOCK DIAGRAM FOR  
HARRY DIAMOND LABORATORIES/ROCK ISLAND ARSENAL,  
GUN STABILIZATION SYSTEM  
MIC-V TANK WITH 20MM GUN

$\omega_n = 49$  RADIANS PER SECOND  
 $\zeta = 1$   
 $\tau_G = 0.7$  SECOND (ACTUATOR AREA = 0.89 SQUARE INCH)  
 $\tau = 2.0$  SECONDS  
 $\tau_3 = 0.01$  SECOND



EFFECT OF LEAD TIME CONSTANT ON  
DISTURBANCE SUPPRESSION OF  
GUN STABILIZATION SYSTEM

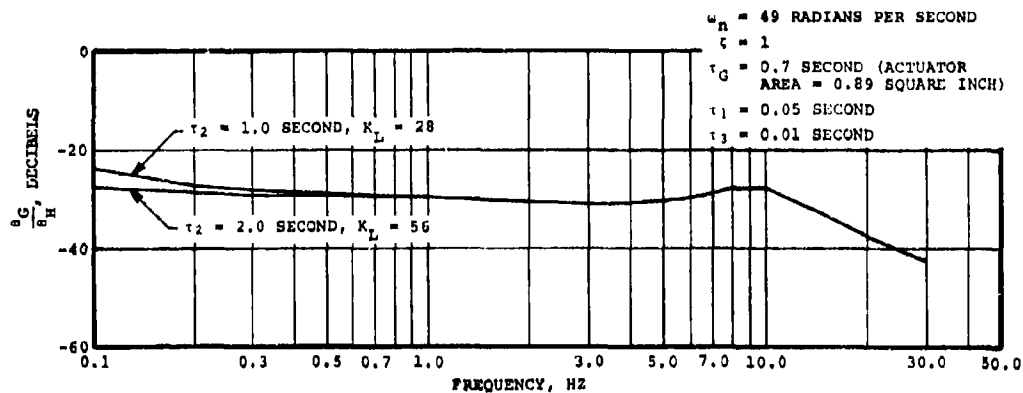


FIGURE 4  
EFFECT OF LAG TIME CONSTANT ON  
DISTURBANCE SUPPRESSION OF  
GUN STABILIZATION SYSTEM

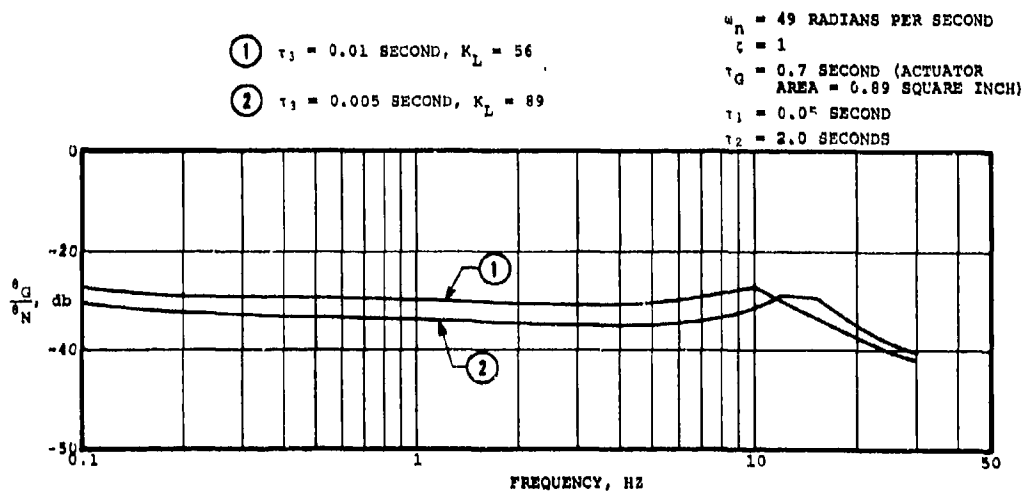


FIGURE 5  
EFFECT OF SERVOVALVE TIME CONSTANT ON  
DISTURBANCE SUPPRESSION OF  
GUN STABILIZATION SYSTEM

Figure 4 illustrates the minimal effect of variations in the lag time constant,  $\tau_2$ . Reduction of  $\tau_2$  from 2.0 seconds to 1.0 second results in degradation only at a very low frequency.

Figure 5 illustrates the effect of the time constant representing the response of the fluidic/hydraulic servovalve interface. These curves indicate that the time constant of the servovalve should be decreased to a value as small as practical within the power limitations of the fluidic circuit.

## 2.2 Controller Noise Analysis

An indication of the behavior of the gun stabilization system in response to noise generated within the controller was obtained by studying the block diagram in Figure 6.

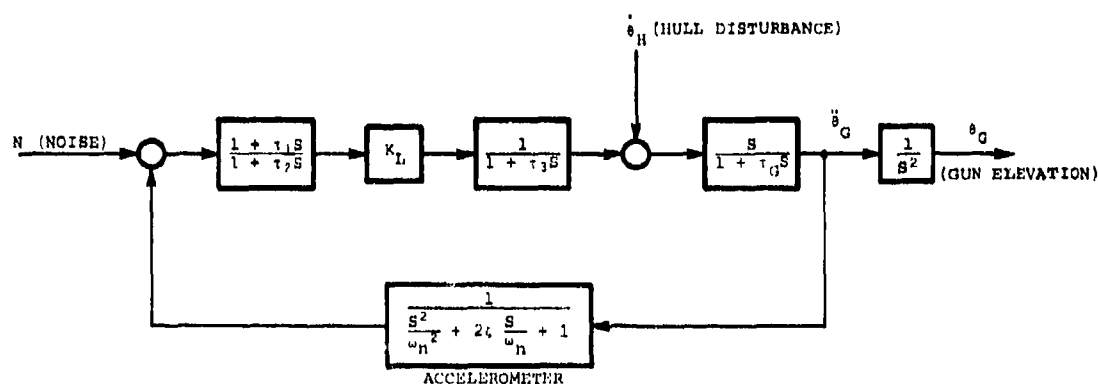


FIGURE 6

### BLOCK DIAGRAM FOR NOISE ANALYSIS

The transfer function of gun elevation for noise input is given by the following:

$$\frac{\theta_G}{N} = \frac{1}{s^2} \frac{G}{1 + GH}$$

where

$$G = \frac{(1 + \tau_1 S) K_L S}{(1 + \tau_2 S)(1 + \tau_3 S)(1 + \tau_G S)}$$

$$H = \frac{1}{\frac{s^2}{\omega_n^2} + 2\zeta \frac{s}{\omega_n} + 1}$$

from which the amplitude of gun motion in response to controller noise is obtained as

$$\left| \frac{\theta_G}{N} \right| = \left| \frac{1}{s^2} \frac{G}{1 + GH} \right|_{s \rightarrow j\omega}.$$

Controller noise amplitude was obtained by observing oscilloscope traces of controller output with the volume cans removed to eliminate the filtering effect of the lag-lead compensator. Noise amplitude distribution with respect to frequency appeared to be approximately Gaussian, with a center frequency of 3.3 Hz, a standard deviation of 1.1 Hz, and a maximum peak-to-peak amplitude of 0.0046 radian per second<sup>2</sup> (0.04 psi assuming

loop gain  $K_L = 55$  and  $\frac{\Delta P_F}{\ddot{\theta}_G} = 0.1572$  psid per radian per second<sup>2</sup>).

Thus

$$|N| = 0.0046 e^{-1/2} \left( \frac{f - 3.3}{1.1} \right)^2$$

so that the amplitude of gun motion in response to controller noise is:

$$|\theta_G| = \left| \frac{\theta_G}{N} \right| |N|$$

$$|\theta_G| = \left| \frac{\theta_G}{N} \right| 0.0046 e^{-1/2} \left( \frac{f - 3.3}{1.1} \right)^2.$$

It is evident from the plot of amplitude as a function of frequency in Figure 7 that, although the system accommodates controller noise very well at the center frequency of 3.3 Hz, the system is sensitive to noise at low frequencies. The effect of this characteristic is to cause a slow drift in response to variations in supply pressure or other disturbances to the system.



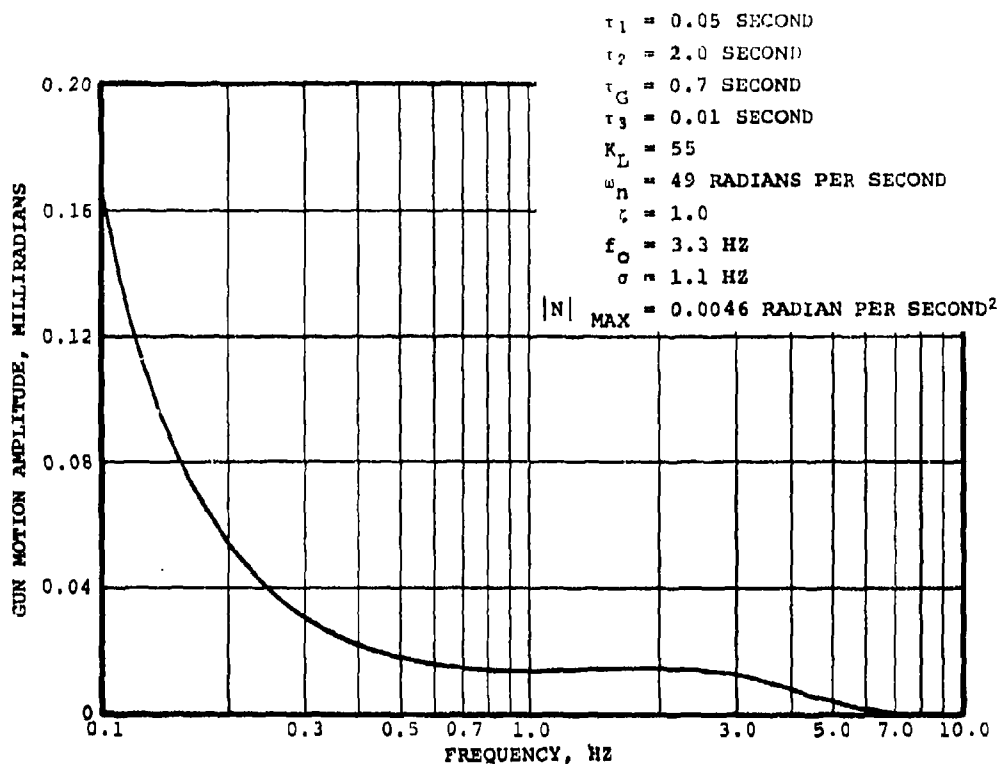


FIGURE 7

EXPECTED GUN MOTION AMPLITUDE  
RESULTING FROM CONTROLLER NOISE

### 2.3 Analog Computer Simulation

The following paragraphs describe the analog computer simulation of the system.

**2.3.1 Method of System Analysis** - To evaluate the effects of accelerometer non-linearities and to determine optimum values of controller parameters, the gun elevation control system was modeled on an analog computer. In an accelerometer-stabilized system, the gunner is considered to be the primary controller, performing the target acquisition and tracking function while the accelerometer loop provides assistance in steadying the gun as the tank hull is being subjected to high frequency motions over the terrain. Since the gunner is an essential part of the control system, his dynamic performance was

included in the analog model. The analog computer circuit used to implement the reduced system block diagram is presented in Figure 8.

The circuit was programmed on the EAI 680 analog computer. The output of the computer was available as a time trace on a multi-channel strip chart recorder. The computer output could also be processed to yield frequency response information by use of a Bafco phase/gain analyzer.

The effects of the following parameters were evaluated:

- $\omega_n$  = Accelerometer bandwidth
- $\zeta$  = Accelerometer damping ratio
- D = Accelerometer threshold
- $A_p$  = Hydraulic actuator piston area

Values of other parameters were fixed as follows:

- $\tau_1$  = 0.1 second
- $\tau_2$  = 2.0 seconds
- $\tau_3$  = 0.01 second
- $\tau_d$  = 0.05 second
- $\omega_H$  = 25 radians per second (4.0 Hz)
- $\zeta_H$  = 0.1

**2.3.2 Results of System Analysis** - Figures 9 through 13 show the effects of variations in several system parameters on the ability of the gun stabilization system to reduce gun motion when the tank hull is traveling at various speeds over a sinusoidal terrain. These curves were obtained by applying a signal representing a sinusoidally varying terrain with an amplitude of 1.6 degrees (0.0279 radian) to the analog model at the  $\theta_B$  input and observing the signal representing gun angle,  $\theta_G$ . The ratios (in decibels) of the amplitudes of gun angle to terrain angle are plotted as a function of terrain frequency. The effectiveness of the stabilization system in suppressing disturbances is measured by the amount that the



curves fall below zero decibels. A value of minus 28.9 db represents a gun motion of 0.001 radian in response to a hull motion of 0.030 radian. In each case, the loop gain,  $K_L$ , was set at a value that gave reasonably stable performance.

The three curves that appear in each of Figures 9, 10, 12, and 13 demonstrate the effect of accelerometer threshold or deadband on the ability of the system to suppress gun motion. As expected, an increased threshold level decreases the effectiveness of the system. A threshold value of 0.003 radian per second<sup>2</sup> is attainable with a 40 radian per second accelerometer of six-inch diameter.

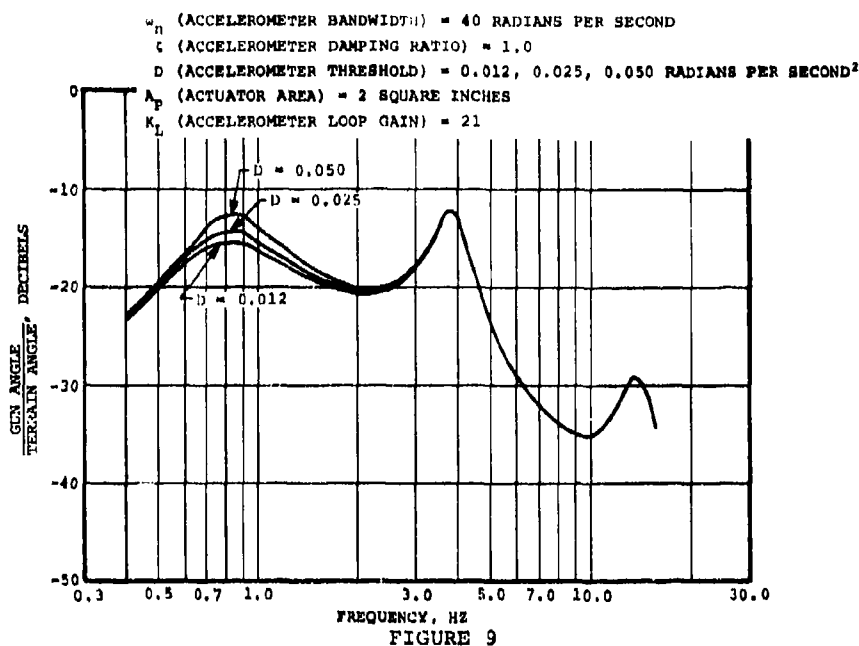
Figure 9 represents the suppression obtainable with the existing actuator with a piston area of 2.0 square inches. The peak occurring at 0.8 Hz represents the boundary between the low frequency range where the gunner provides suppression and the high frequency range where the gun stabilization system provides suppression. The peak occurring at 4 Hz is caused by the underdamped hull suspension system going into resonance, resulting in oscillation of the tank hull at a greater amplitude than terrain oscillation. Performance exhibited by this case is poor since the suppression obtainable with a threshold of 0.012 radian per second is only minus 15 db at a frequency of 0.8 Hz. It is noted that doubling of the accelerometer bandwidth as shown in Figure 10 is of no value in improving suppression of disturbances.

Figure 11 represents the disturbance suppression obtainable when the actuator piston area,  $A_p$ , is decreased to 0.89 square inch. The improved suppression is a result of decreased hull coupling as reflected in the increased value of  $\tau_G$ .

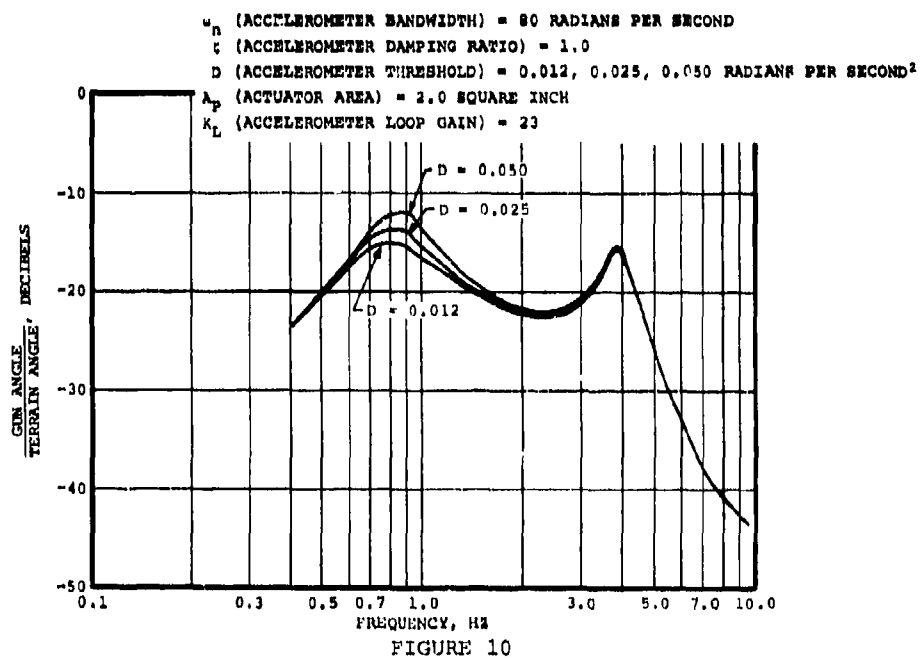
Figure 12 shows the effect of varying the accelerometer damping ratio,  $\zeta$ . These curves indicate that a higher loop gain,  $K_L$ , and therefore better suppression, is attainable when the damping ratio is fairly high.

Figure 13 also indicates that increased accelerometer bandwidth is of no value when the actuator piston area is decreased.

Figure 14, which is included for comparison purposes, indicates the disturbance suppression obtained when the accelerometer gain,  $K_L$ , is set at zero. This represents the suppression provided by the gunner alone when disturbances are attenuated only by the dynamics of the hull suspension and hydraulic actuation system.



#### DISTURBANCE SUPPRESSION WITH GUN STABILIZATION SYSTEM



#### DISTURBANCE SUPPRESSION WITH GUN STABILIZATION SYSTEM

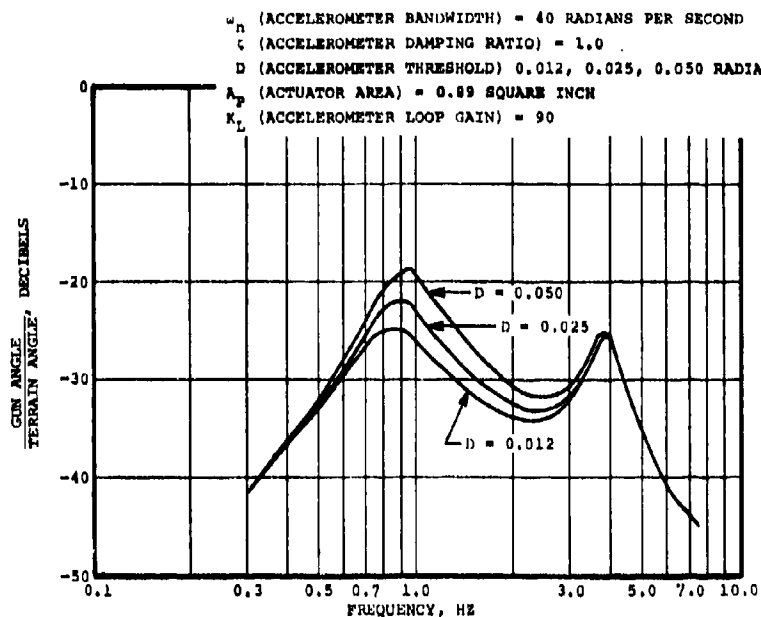


FIGURE 11

# DISTURBANCE SUPPRESSION WITH GUN STABILIZATION SYSTEM

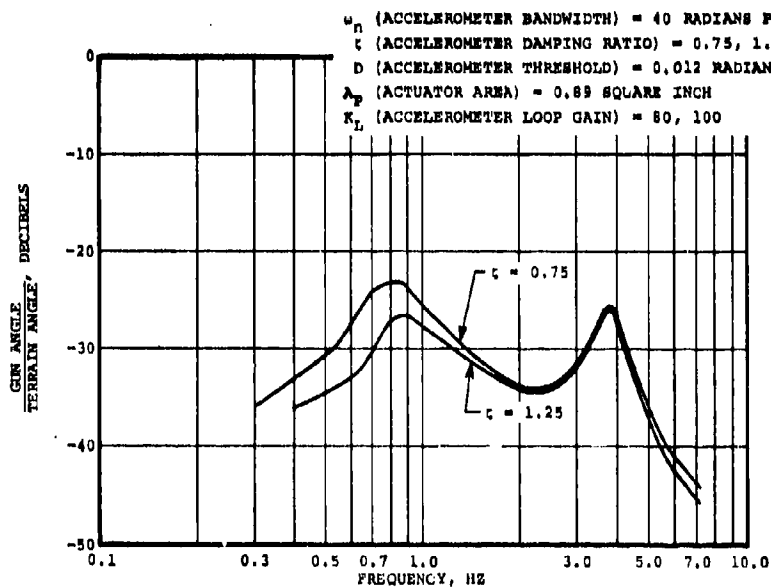


FIGURE 12

# DISTURBANCE SUPPRESSION WITH GUN STABILIZATION SYSTEM

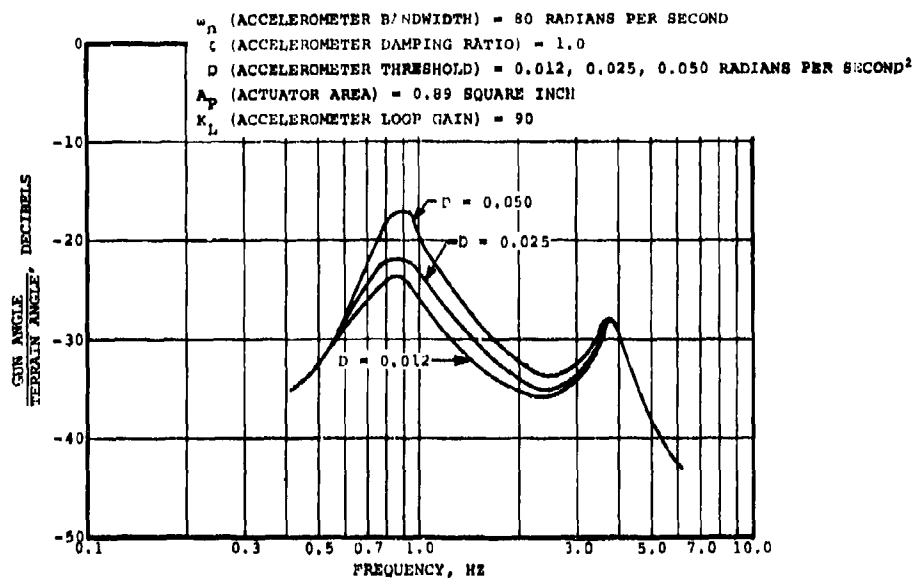


FIGURE 13

# DISTURBANCE SUPPRESSION WITH GUN STABILIZATION SYSTEM

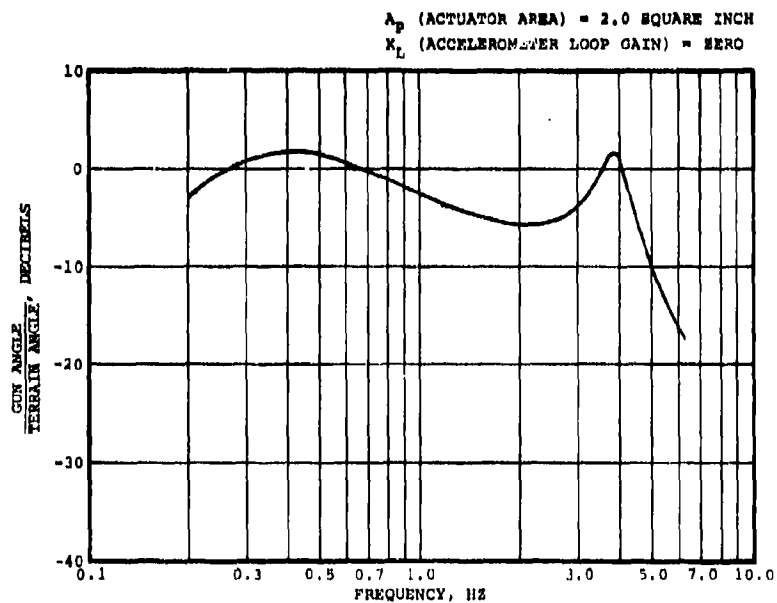


FIGURE 14

# DISTURBANCE SUPPRESSION WITHOUT GUN STABILIZATION SYSTEM (GUNNER ONLY)

Figures 15 and 16 show suppression of gun motion in the time domain in response to sinusoidal and step disturbances in the terrain. The effect of the underdamped hull suspension system in amplifying hull motion at frequencies near its resonance point is noted. It is also noted that the system enters a one Hz limit cycle when subjected to low frequency or no terrain disturbances. This is the primary detrimental effect of accelerometer threshold. If the threshold is increased to 0.05 radian per second<sup>2</sup>, the limit cycle amplitude increases to 0.003 radian.

A comparison of the gun motion in response to a step change in target position, as exhibited in Figure 17, to the gun motion in Figure 16 demonstrates the necessity of the derivative gunner handle pickoff. Without this handle rate signal to suppress the accelerometer feedback signal, the gunner finds it very difficult to change the gun rate to seek a new target position, and gun response is very slow and sluggish.

**2.3.3 Recommendations Based on Analysis** - The parameters most affecting suppression of terrain disturbances appear to be accelerometer threshold and actuator time constant. Suppression is not improved by increasing accelerometer bandwidth beyond that necessary to suppress disturbances at the natural frequency of the hull suspension system.

Because of the relation between bandwidth and threshold, it is recommended that the design bandwidth be set at a value of approximately 40 radians per second. Because of the effect of actuator piston area on disturbance suppression, it is recommended that the piston area be reduced to 0.89 square inch. This results in a value of  $\tau_G$  of 0.7 second.

It is recommended that a handle rate pickoff be included in the system. The response of the pneumatic/hydraulic servo-valve should be less than 0.01 second. Accelerometer damping should be at least critical.

### 3. CONTROLLER DESIGN

A schematic diagram of the controller circuit is presented in Figure 18. The controller utilizes a pneumatic medium for sensing and processing inputs from the gunner's handle and from the accelerometer. The resulting pneumatic output is converted to a hydraulic signal by the pneumatic/hydraulic servovalve. The components of the controller furnished by AiResearch were the accelerometer, handle rate pickoff, and fluidic signal processing circuit. The gunner's handle, servovalve, and actuator were furnished by the U.S. Army (Rock Island Arsenal).

Photographs of the assembled controller and gunner's handle are presented in Figures 19, 20, and 21.



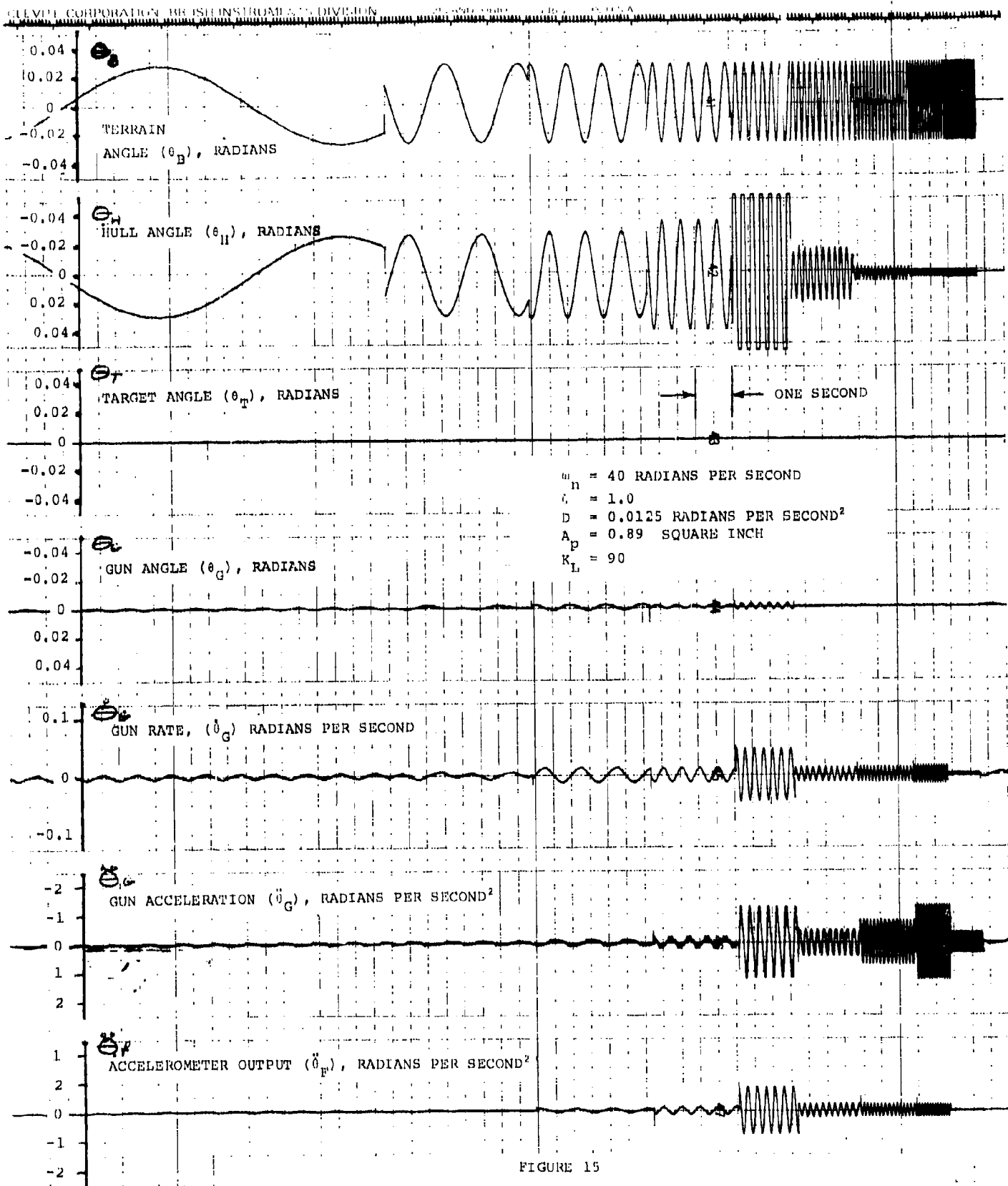


FIGURE 15

TIME RESPONSE TO SINUSOIDAL TERRAIN

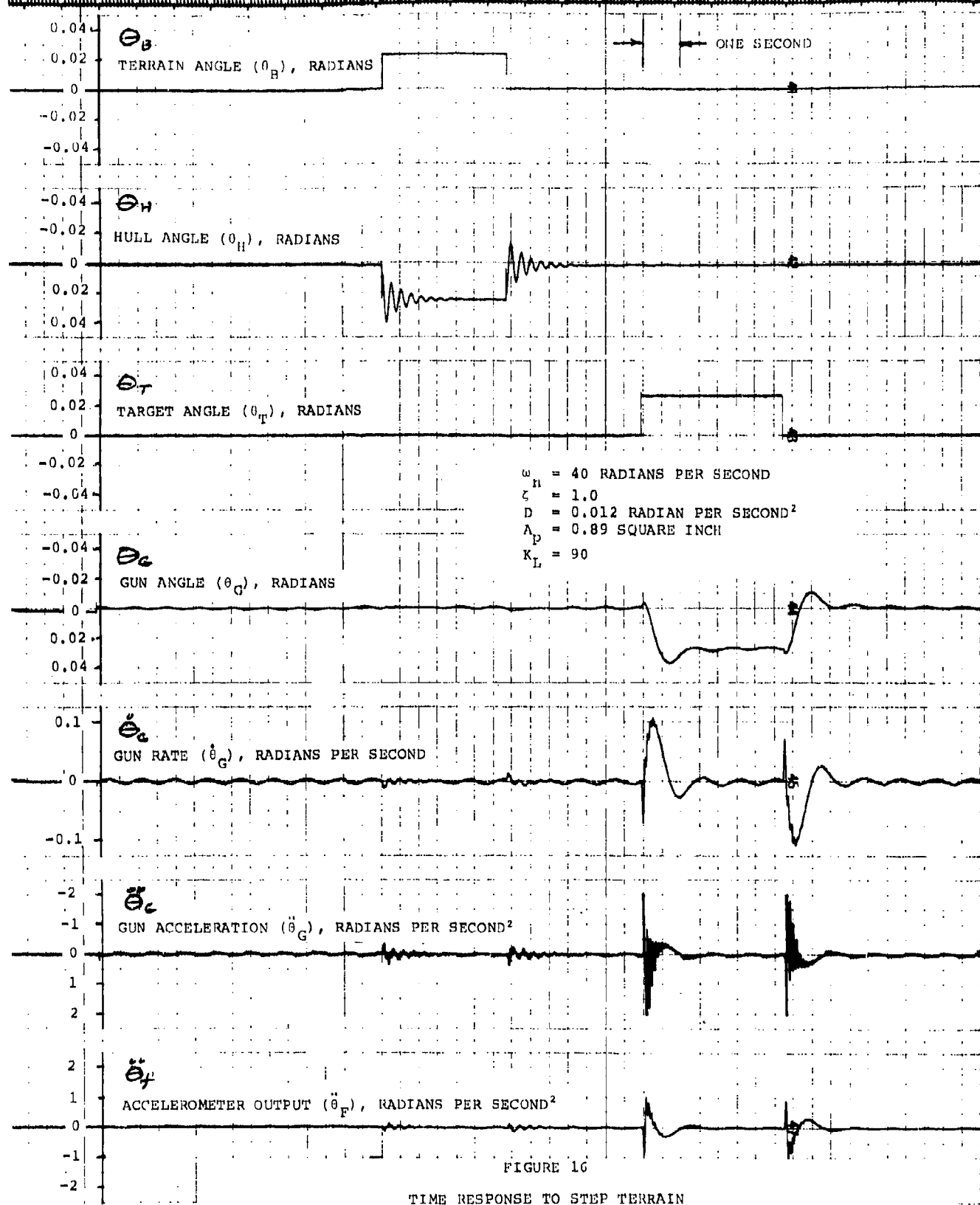


FIGURE 16  
TIME RESPONSE TO STEP TERRAIN  
AND STEP TARGET

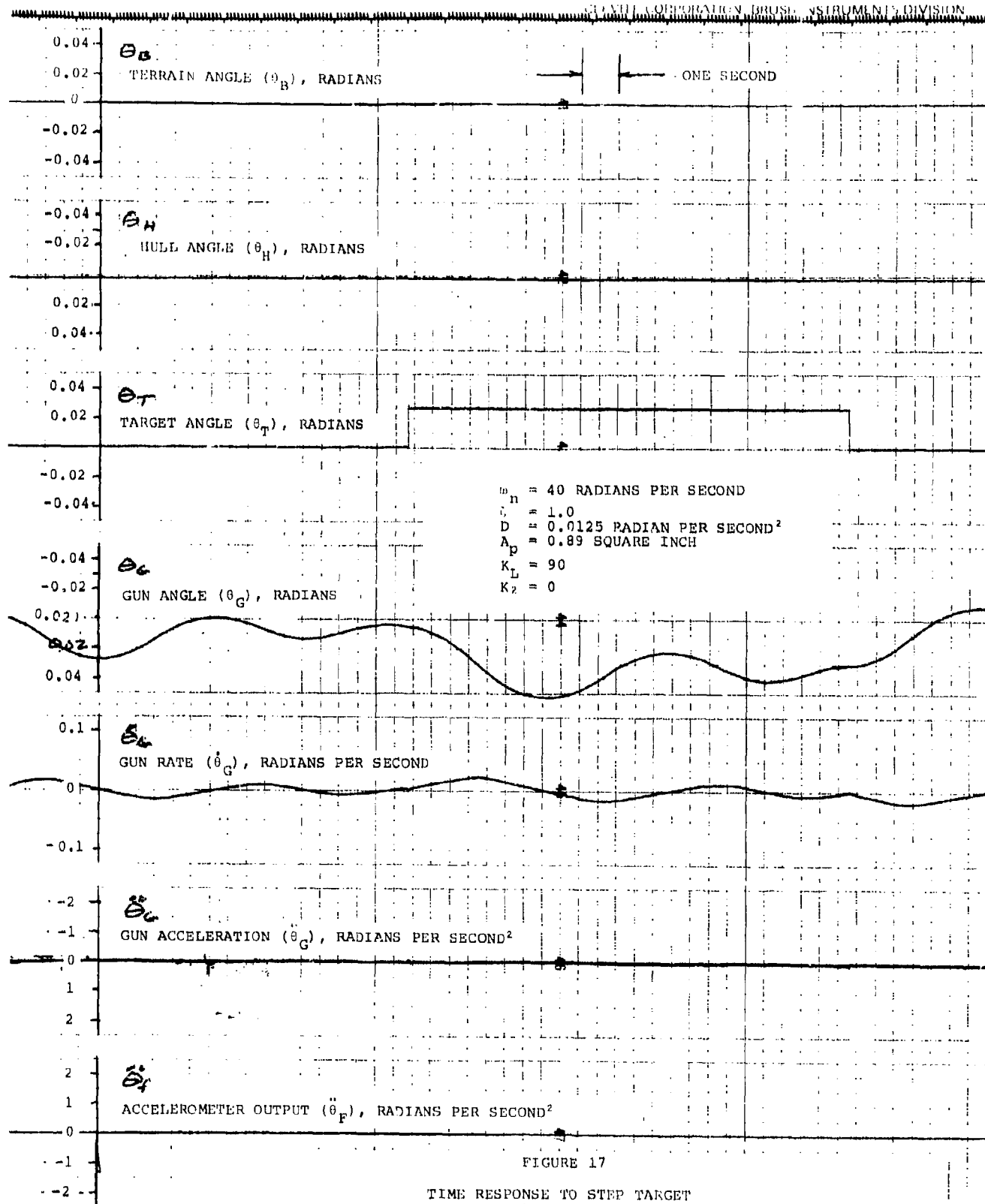


FIGURE 17

 TIME RESPONSE TO STEP TARGET  
 WITHOUT DERIVATIVE HANDLE PICKOFF

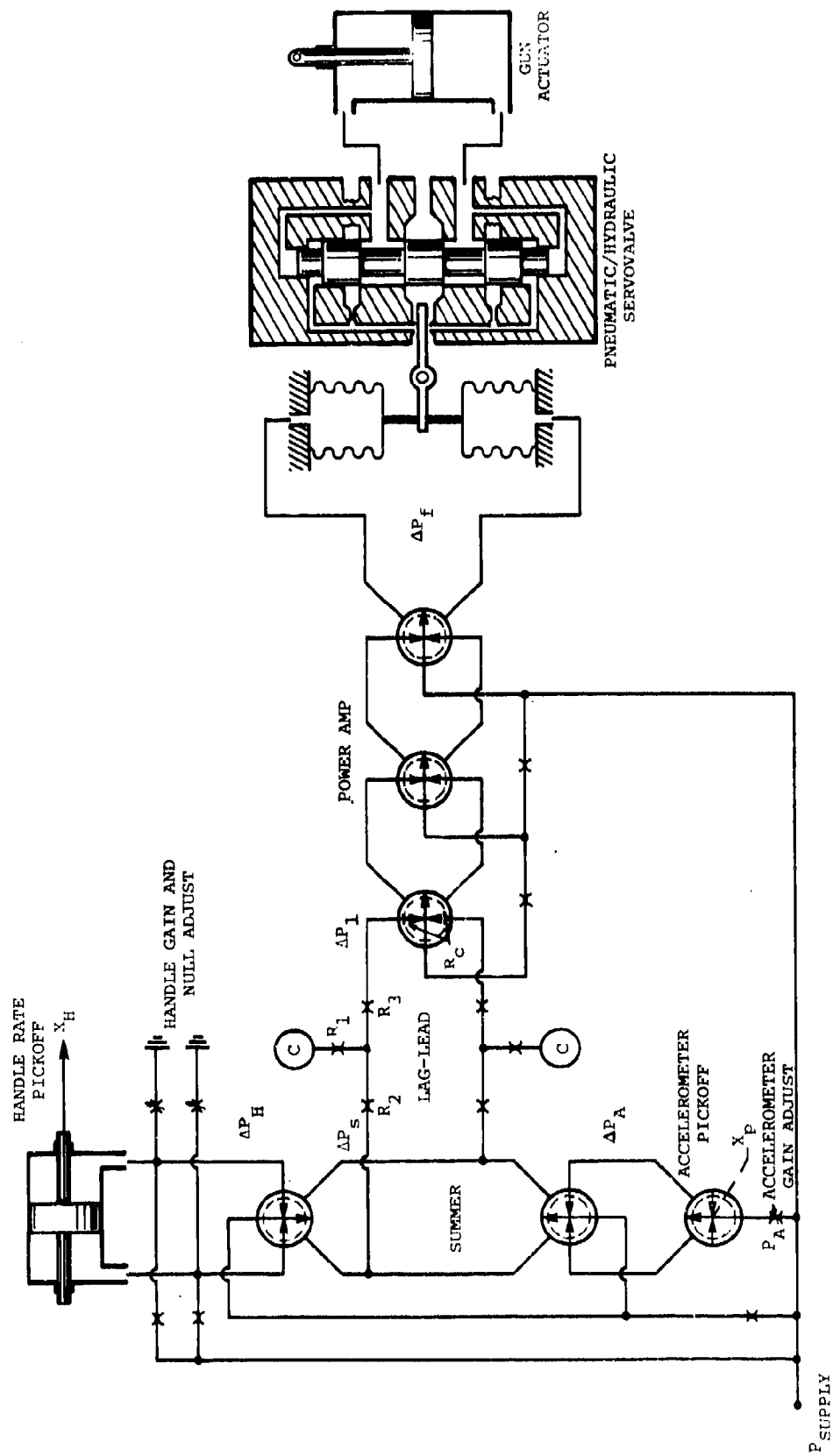
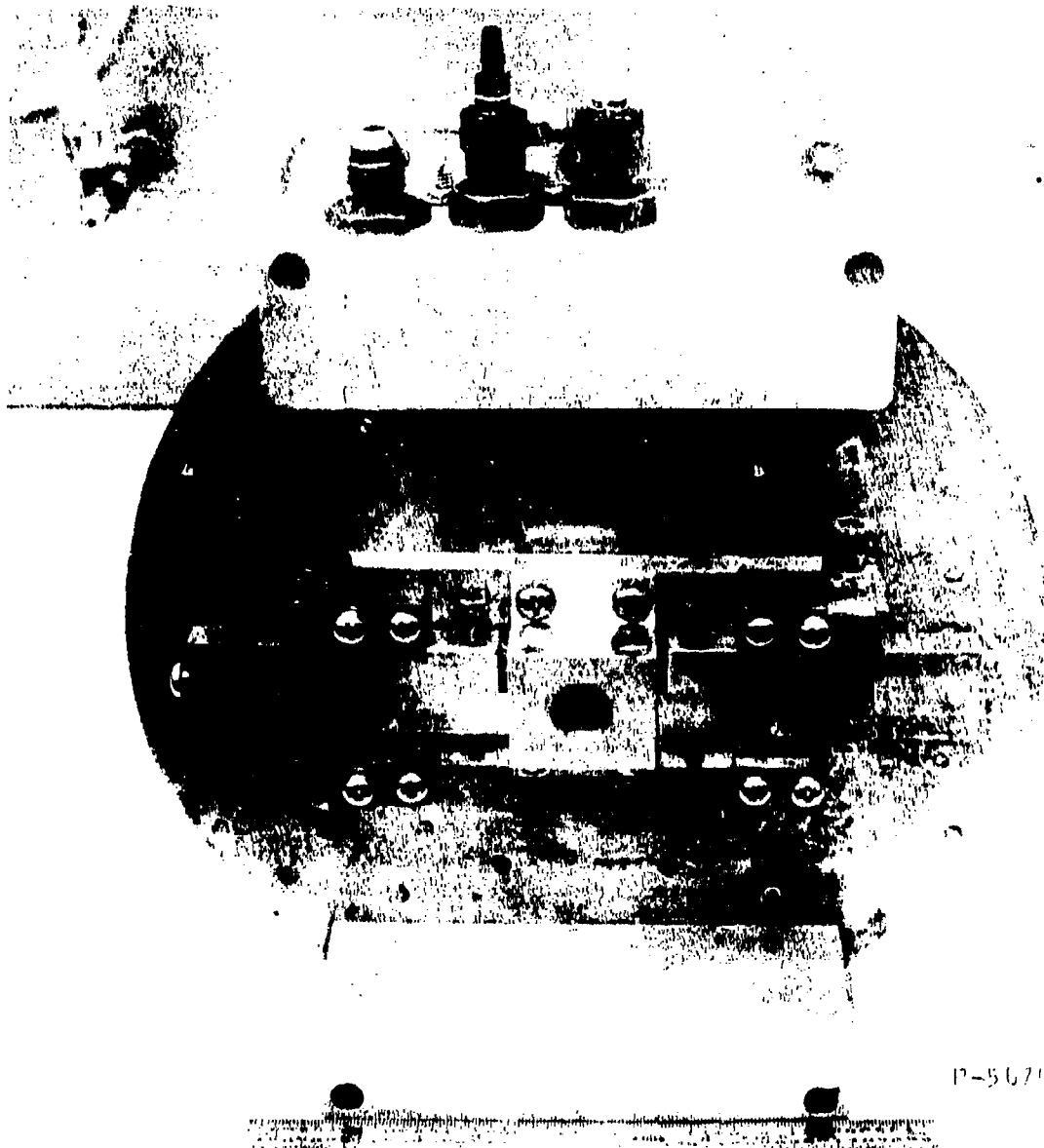


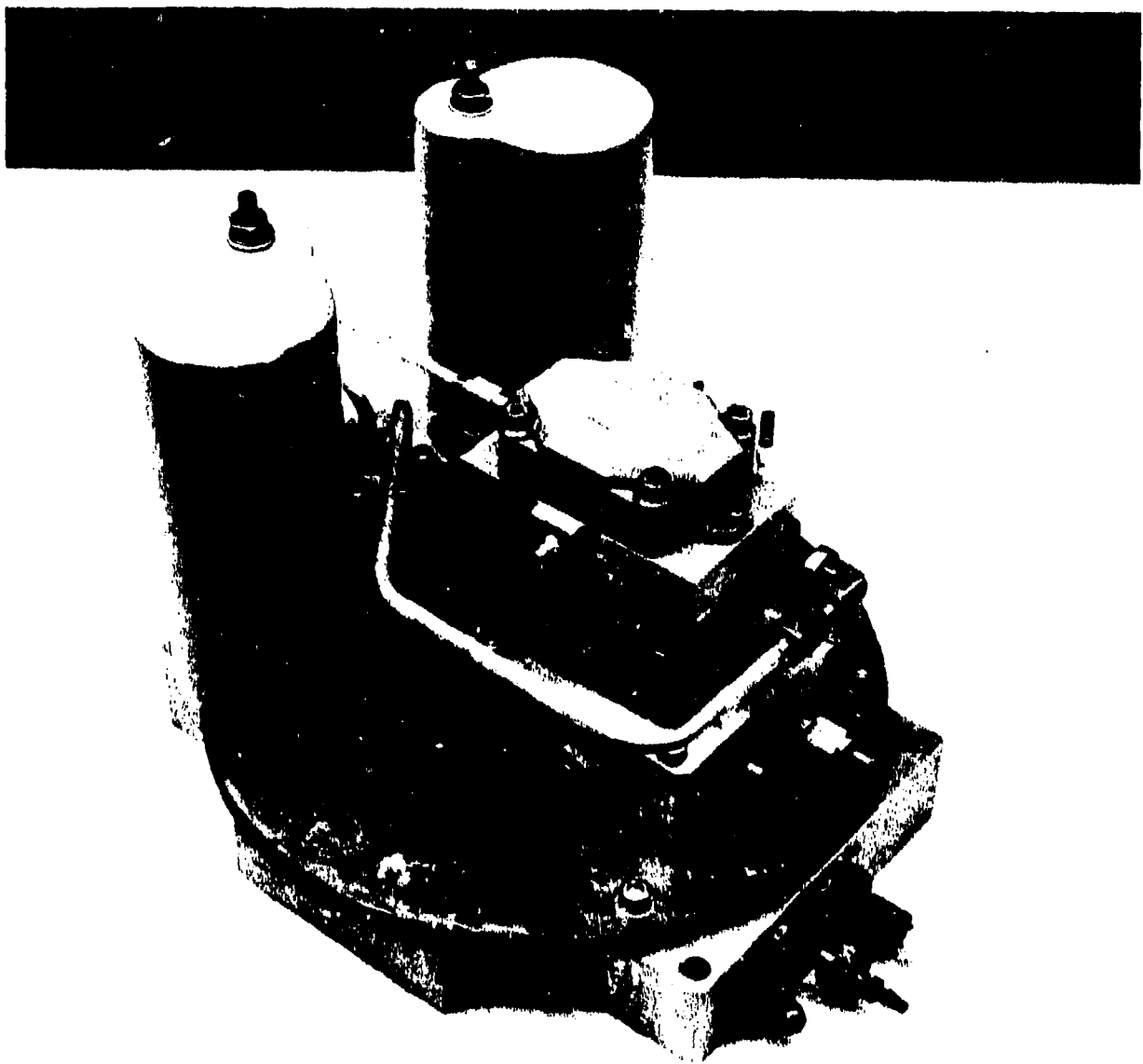
FIGURE 18  
FLUIDIC CONTROLLER CIRCUIT



P-5670 -5

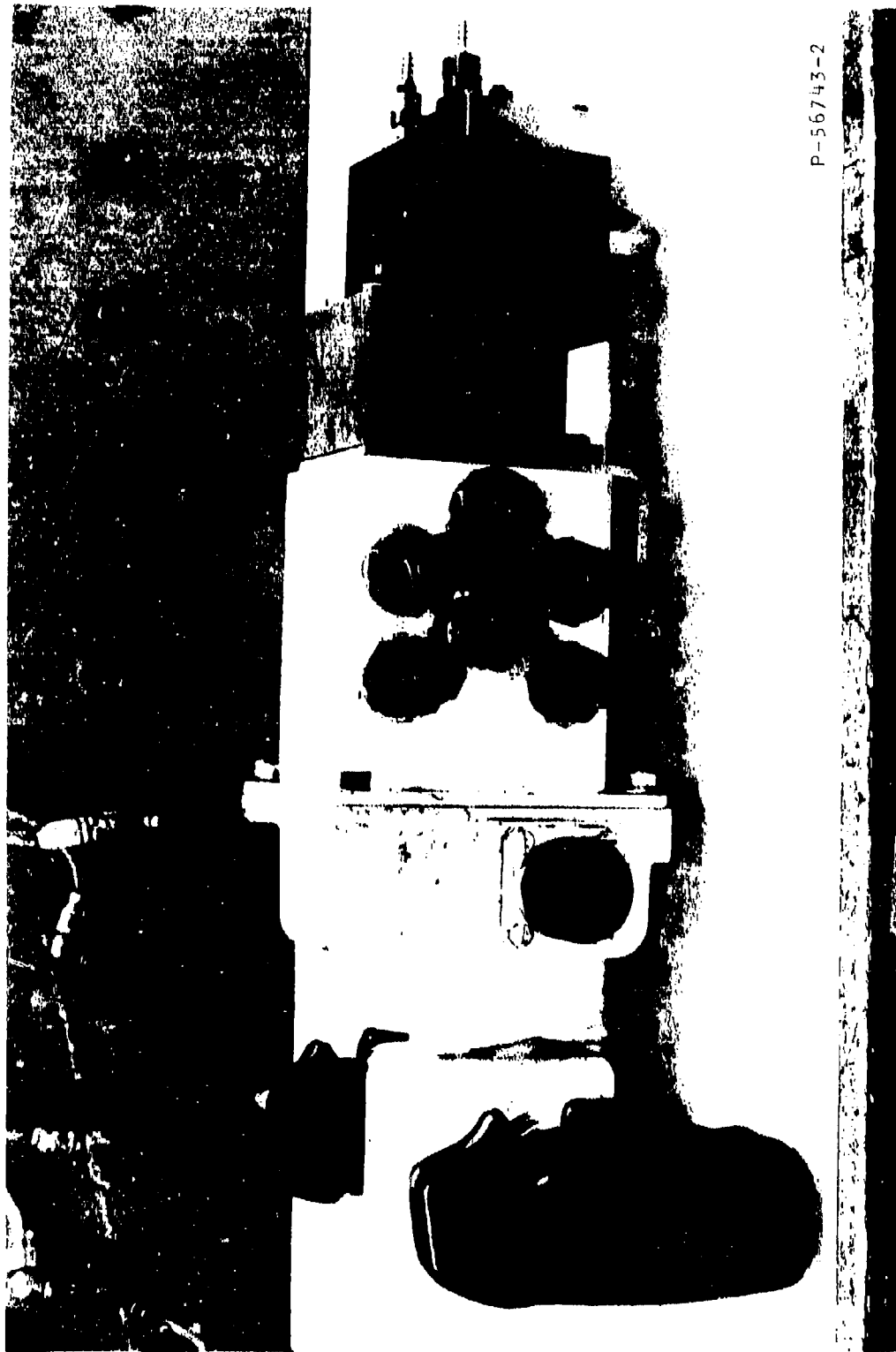
FIGURE 19

HDL FLUIDIC GUN STABILIZATION SYSTEM  
(ACCELEROMETER SIDE)



P-56743-3

FIGURE 20  
HDL FLUIDIC GUN STABILIZATION SYSTEM  
(CONTROLLER SIDE)



P-56743-2

FIGURE 21

HANDLE VALVE MODIFICATION  
HDL FLUIDIC GUN STABILIZATION SYSTEM

### 3.1 Controller Equation

To obtain a practical specification for the accelerometer-controller unit, the block diagram shown in Figure 1 was reduced by the steps shown in Figure 22. The final block diagram represents the fluidic accelerometer-controller unit developed by AiResearch. The unit receives the inputs of gun acceleration,  $\ddot{\theta}_G$ , and handle valve position,  $X_H$ , and produces a pneumatic output,  $\Delta P_f$ , to be applied to the servovalve. Using the values listed in Figure 1, the input-output relation for the device is determined from the block diagram to be as follows:

$$\Delta P_f = \left[ 0.1572 K_L \frac{S^2 \ddot{\theta}_G}{\frac{S^2}{49} + \frac{25}{49} + 1} + 0.3743 K_L \frac{S X_H}{1 + 0.05S} \right] \times \left[ \frac{1 + 0.05S}{(1 + 2S)(1 + 0.01S)} \right]$$

As outlined in Appendix B, the accelerometer transfer function is computed to be

$$X_P = \frac{r}{\omega_n^2} \frac{\ddot{\theta}_G}{\frac{S^2}{\omega_n^2} + 2 \frac{\zeta S}{\omega_n} + 1}$$

$$X_P = (1.25 \times 10^{-3}) \frac{S^2 \ddot{\theta}_G}{\frac{S^2}{\omega_n^2} + 2 \frac{\zeta S}{\omega_n} + 1} \text{ inches,}$$

where  $X_P$  is the displacement input to the pin amplifier pickoff.

The equation describing the fluidic controller circuit is then:

$$\Delta P_f = \left( 125.76 K_L X_P + 0.3743 K_L \frac{S X_H}{1 + 0.05S} \right) \times \left( \frac{1 + 0.05S}{(1 + 2S)(1 + 0.01S)} \right) \text{ psid.}$$



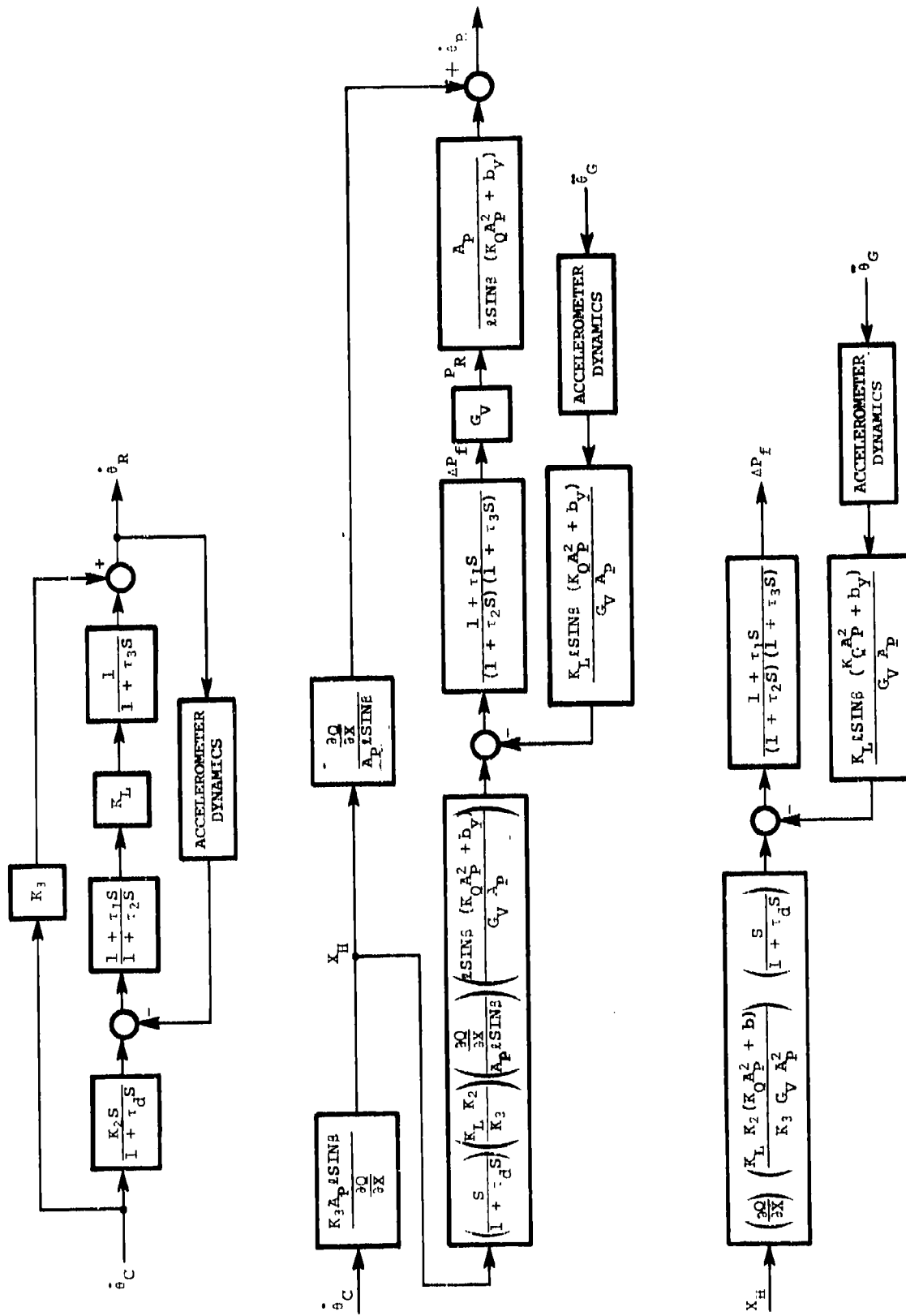


FIGURE 22  
DERIVATION OF ACCELEROMETER CONTROLLER  
BLOCK DIAGRAM

### 3.2 Accelerometer

As shown in Figure 23, the accelerometer consists of a rectangular proof mass suspended at the center of mass by a rotational spring flexure. Motion of the mass relative to its housing is sensed by a pin amplifier which measures the relative displacement of the end of the proof mass. The mass is provided with balancing screws to place the center of mass precisely at the center of rotation. This minimizes the sensitivity of the accelerometer to linear accelerations.

The pin amplifier, in its present state of development, has a resolution of approximately  $2 \times 10^{-6}$  inches. This corresponds to an acceleration resolution of  $1.71 \times 10^{-3}$  radians per second<sup>2</sup>, which is considered adequate for suppressing gun motions to an amplitude approximately 25 db down from hull disturbance inputs.

Motion of the proof mass is damped by a magnetic fluid suspended by a pair of cylindrical magnetic pole pieces protruding into cylindrical cavities in the proof mass, forming a squeeze film. This mechanism has been found to provide an excellent method of damping for breadboard and feasibility demonstration components, primarily because of its adjustability. The nature of the presently available diester base magnetic fluid may preclude its use in a production system unless further improvements are made in this fluid. Two properties of this fluid cause concern: (a) the temperature-viscosity properties may limit the temperature range over which system operation is satisfactory, and (b) the volatility of the fluid will limit the operational and shelf life of the system. For these reasons, alternate methods of accelerometer damping are required.

The rotational spring flexure which was used in the breadboard units is a commercially available device commonly used in many instrument designs. Information which is available on this flexure design indicates that cross-axis sensitivity of the accelerometer could be improved by substitution of a cruciform design flexure. The cruciform design was developed by AiResearch for an electrofluidic transducer designed for use in a severe shock environment. The cruciform flexure exhibits the high cantilever/rotational spring rate ratio necessary for reduction of cross-axis sensitivity. In addition, there is greater design flexibility and a significant cost advantage in the use of this flexure. The cruciform design will be investigated further for any two-axis stability system that may be developed.

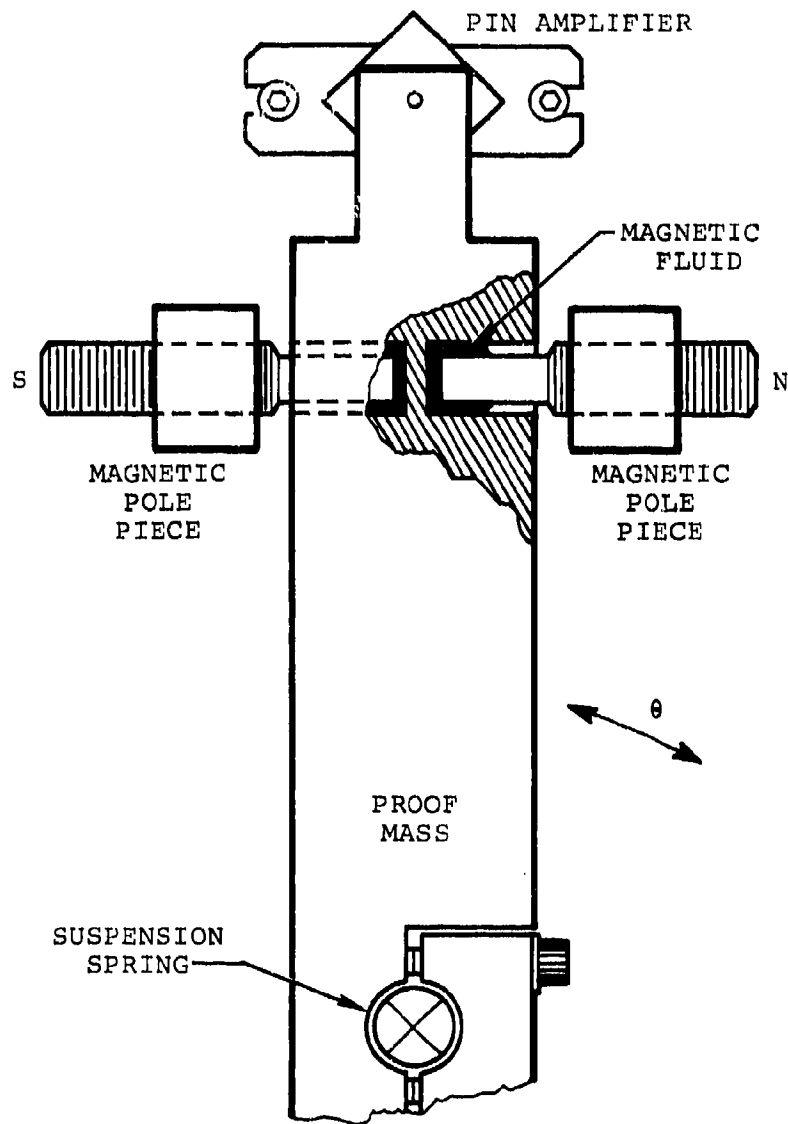


FIGURE 23  
FLUIDIC ACCELEROMETER FOR  
GUN STABILIZATION SYSTEM

### 3.3 Fluidic Controller Circuit

The controller circuit was built in accordance with the schematic diagram in Figure 18. The circuit operates with a supply pressure of 2.0 psig and the output pressure,  $\Delta P_f$ , has a range of  $\pm 1.0$  psid.

The lag-lead dynamic compensation is performed by the resistance-capacitance network and is governed by the relation

$$\frac{\Delta P_\ell}{\Delta P_S} = \left( \frac{R_C}{R_2 + R_3 + R_C} \right) \left[ \frac{1 + R_1 CS}{1 + \left( R_1 + \frac{R_2 R_3 + R_2 R_C}{R_2 + R_3 + R_C} \right) (CS)} \right]$$

A lag time constant of approximately 2.0 seconds was obtained with a volume of 17 cubic inches and four laminations of capillary resistor (AiResearch Part 3155026) serving as  $R_2$  and  $R_3$ . The resistance,  $R_1$ , needed to obtain a lead time constant of 0.05 second is the approximate resistance of the transfer passage to the volumes through the fluidic stack.

The handle rate pickoff signal is summed with the accelerometer pin amplifier signal by means of impedance summing, utilizing the output impedances of the amplifiers.

After the summed signal is processed by the lag-lead network, it is amplified by a three-stage cascade to the required pressure gain and to the required power level to drive the input bellows of the servovalve. Amplifiers of a new HDL design, AiResearch Part 3155131, which is superior in both noise and gain characteristics, were used throughout.

### 3.4 Handle Rate Pickoff

The handle rate signal was obtained from a small double-acting piston which was installed on the handle valve. The piston shaft was spring-loaded against the end of the valve spool so that the piston moved with the spool. The chambers on either side of the piston are connected to the control ports of a fluidic amplifier (refer to Figure 18). The relation between the piston displacement,  $X_H$ , and the amplifier input differential,  $\Delta P_H$ , is given by the expression on the following page.

$$\Delta P_H = \left( \frac{2 R A_H \bar{P}_H}{R_g T} \right) \left( \frac{S}{R C_H S + 1} \right) (X_H)$$

where

R is the combined resistance of the amplifier control ports and nulling orifices,

A<sub>H</sub> is the piston area,

P<sub>H</sub> is the average absolute pressure in the piston chambers,

C<sub>H</sub> is the pneumatic capacitance of the piston chambers,

X<sub>H</sub> is the piston displacement,

R<sub>g</sub> is the gas constant for air, and

T is the absolute air temperature.

#### 4. TEST PROGRAM

The tests conducted on the fluidic gun stabilization system and components included the following:

- Bench testing of controller and components
- Determination of open loop dynamic characteristics
- Closed loop disturbance suppression

The following paragraphs describe each of these test areas.

##### 4.1 Bench Testing

4.1.1 Gunner's Handle Valve - A flow test was conducted on the handle valve to determine flow area versus valve displacement. This information was needed to properly scale the handle rate

pickoff signal. The valve was connected as shown in Figure 24 and supplied with MIL-H-5606 hydraulic oil at a pressure of 500 psi. This was the maximum capability of the bench pump at the required flow rate. The flow rate was measured downstream of the valve before entering the sump, using a turbine flowmeter with digital readout.

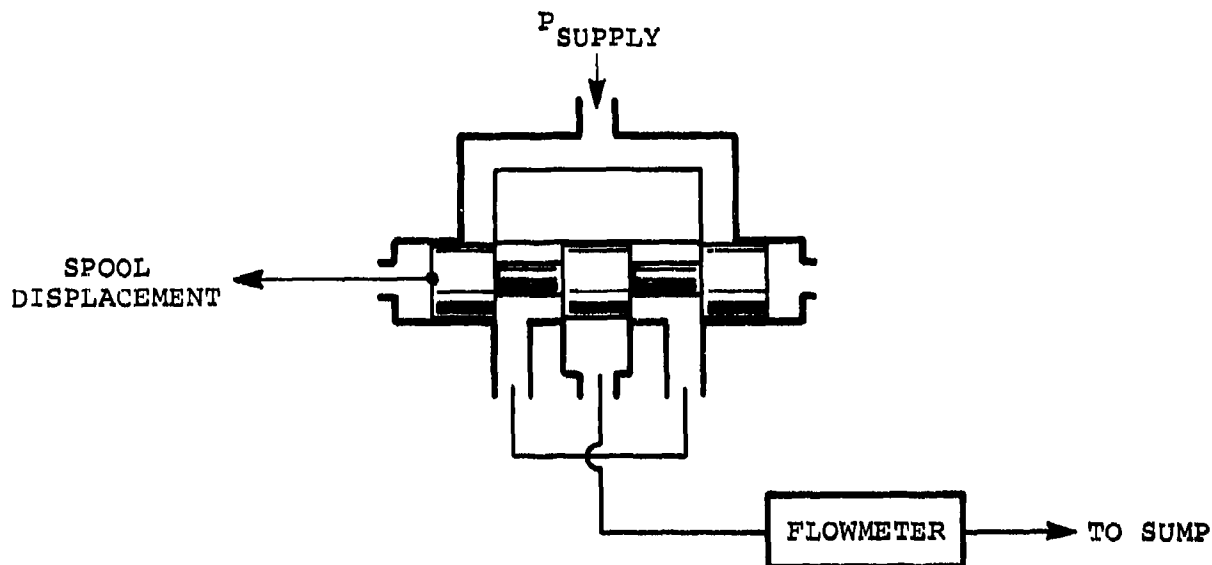


FIGURE 24  
TEST SETUP FOR  
HANDLE VALVE FLOW TEST

The results of the flow test are plotted in Figure 25. Since flow values between  $\pm 0.05$  gpm could not be measured with the available flowmeter, the shape of the curve as it passes through the zero point is estimated.

4.1.2 Accelerometer-Controller Circuit - Dynamic testing of the accelerometer was performed on an oscillating angular rate table which was specially constructed for this program. The table, shown in Figure 26, was capable of oscillating at amplitudes of from 0.005 to 1.0 radian per second over a frequency range from zero to 15 Hz.

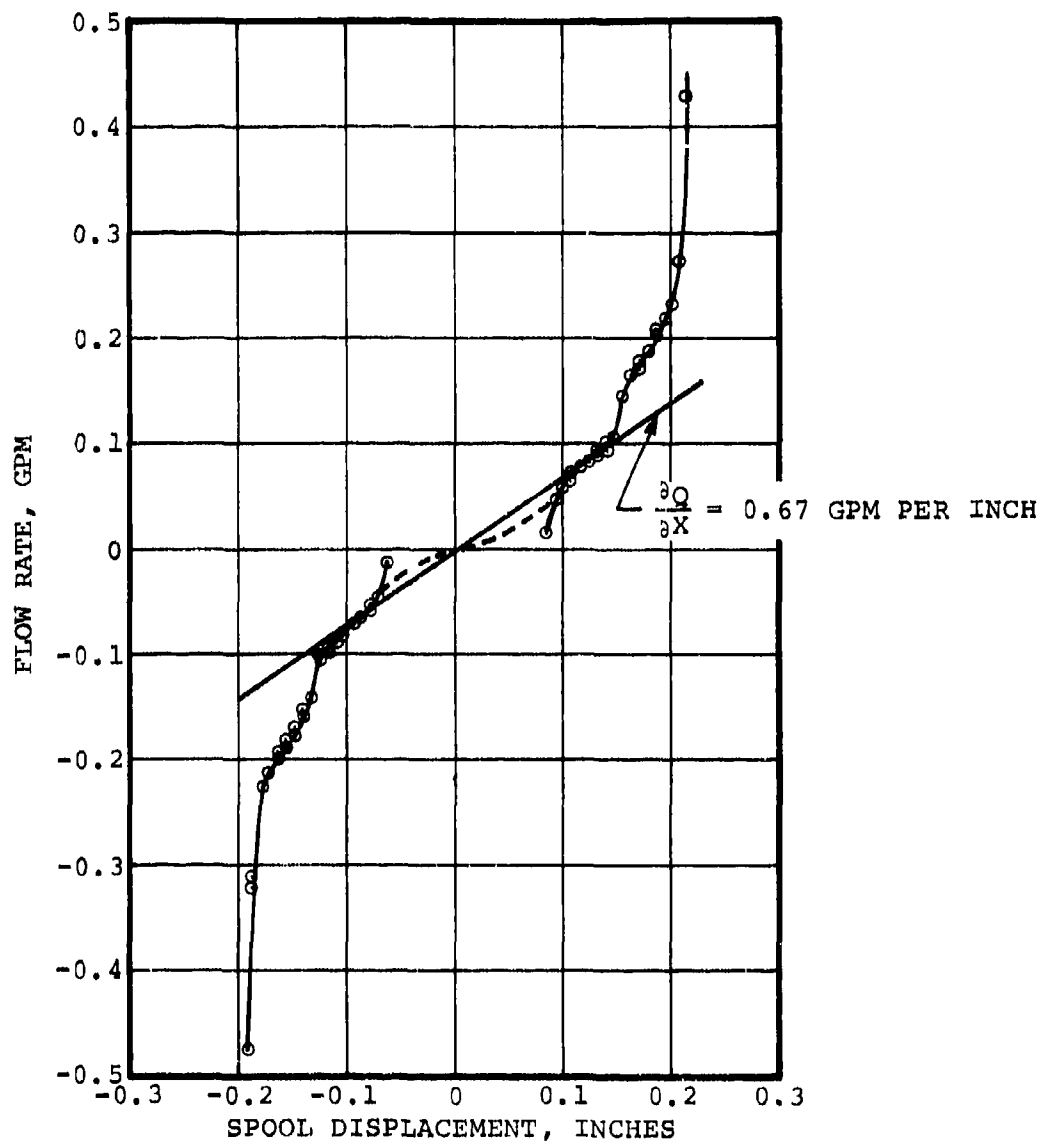
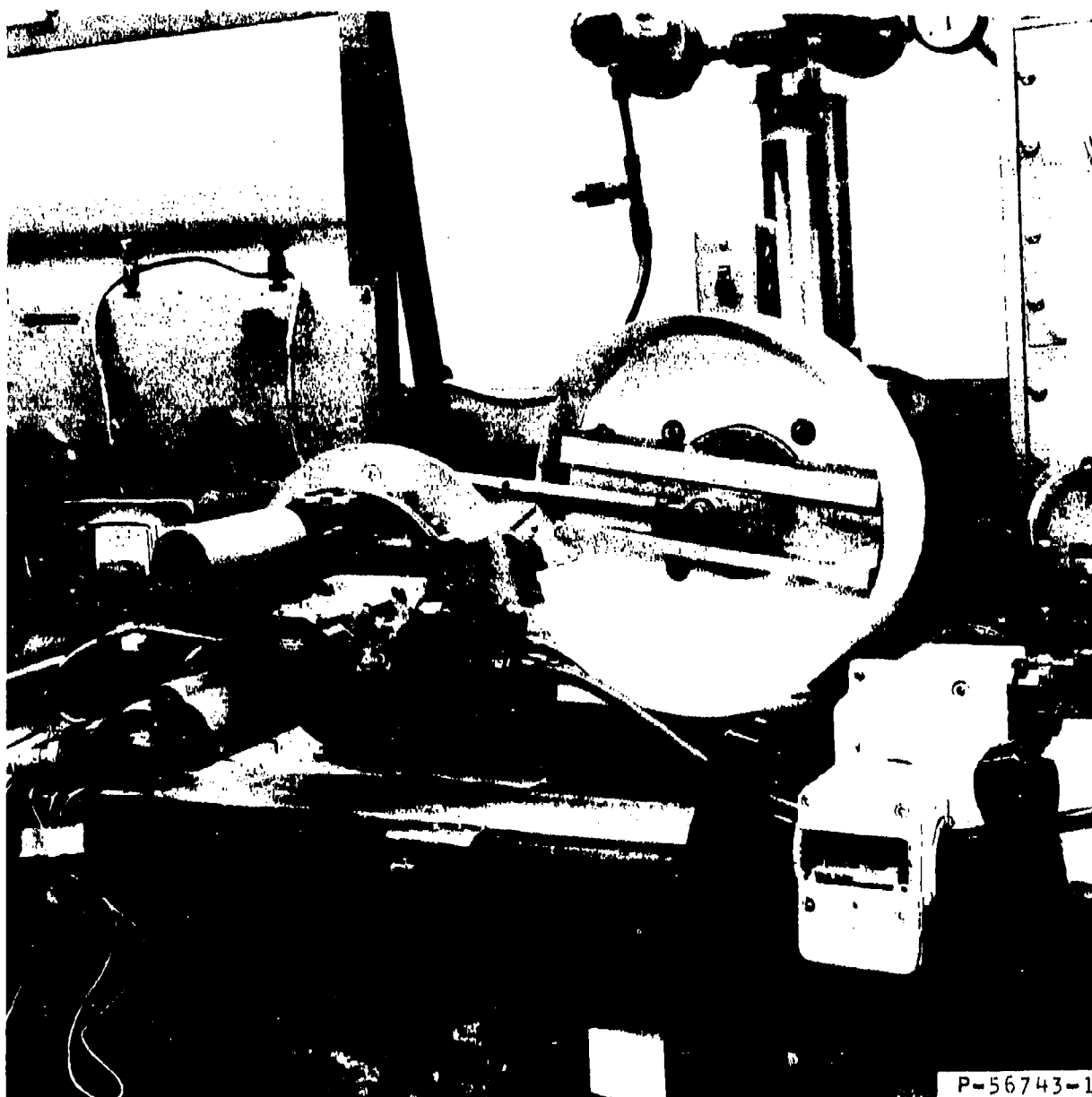


FIGURE 25  
HANDLE VALVE FLOW  
VERSUS SPOOL POSITION



OSCILLATING ANGULAR RATE TABLE  
HDL FLUIDIC GUN STABILIZATION SYSTEM PROGRAM  
FIGURE 26



Difficulty was encountered during dynamic testing of the accelerometer over the full frequency range because of rumble and backlash in the test fixture. Valid data was extracted only at low frequencies and at fairly large oscillation amplitudes. The data obtained, together with the plotted design performance, is presented in Figure 27. It is noted that good agreement was obtained when the oscillation amplitude,  $|\theta_G|$ , was set at 0.1 radian.

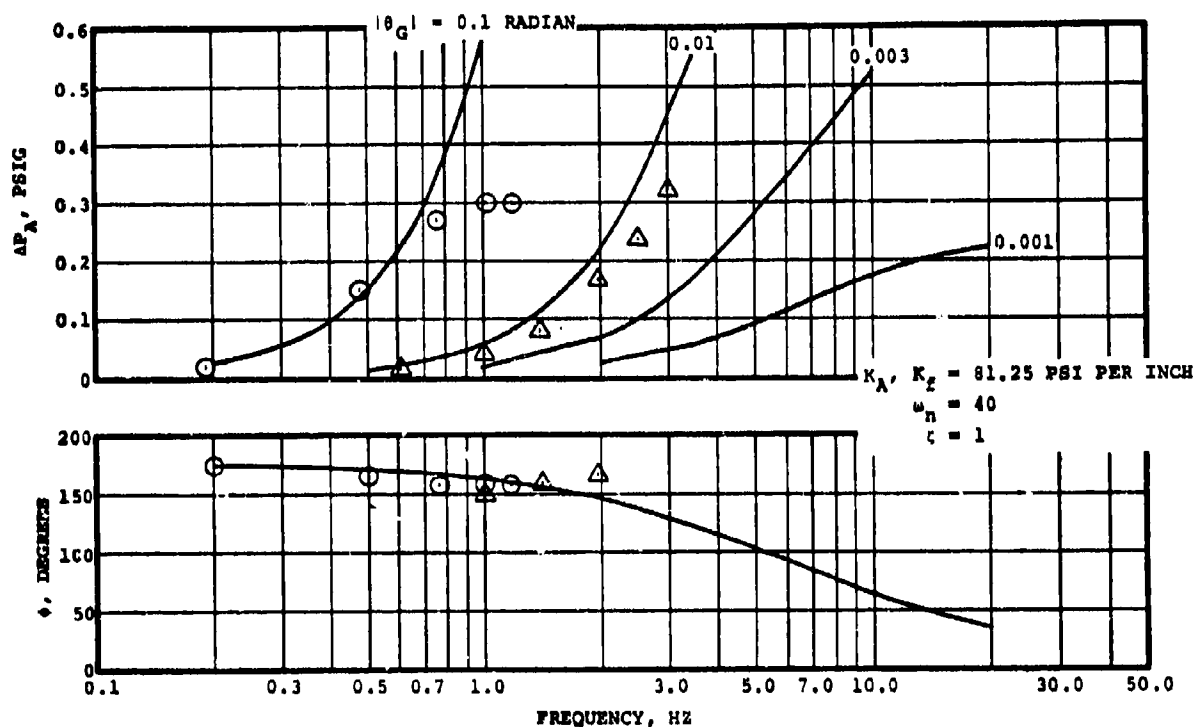


FIGURE 27

PERFORMANCE OF ACCELEROMETER  
FOR GUN STABILIZATION SYSTEM

To circumvent the necessity for testing the controller on the accelerometer test fixture at high frequencies, dynamic testing of the controller circuit was performed by simulating accelerometer inputs with an electrofluidic transducer which contained a pin amplifier identical to the accelerometer pickoff amplifier. The amplitude ratio and phase relationships between fluidic controller output pressure and electrofluidic transducer input current are plotted in Figure 28.

Static performance of the controller circuit was verified by placing the pin amplifier on a fixture which was capable of moving the pin and precisely measuring its position. The output pressure of the controller is plotted versus pin position in Figure 29. Gain of the controller is adjusted by varying the supply pressure to the pin amplifier.

As a check on controller performance and to verify correct operation of the pneumatic/hydraulic servovalve, the controller/servovalve combination was tested on the hydraulic bench. Accelerometer inputs were again simulated with the electrofluidic transducer. The resulting static and dynamic data are plotted in Figures 30 and 31, respectively.

#### 4.2 Open Loop Testing

An open loop frequency response test was performed on the fluidic gun stabilization system to verify the values of system parameters that were used during the analytical phase of the program. The actuation system subjected to test was that installed in the MIC-V turret test fixture located at Rock Island Arsenal. The actuator was powered by the pneumatic/hydraulic pressure control servovalve supplied by the U.S. Army for the program.

As a result of the recommendations presented in paragraph 2.3.3 that the value of  $\tau_G$  be increased to 0.7 second, the actuator piston area was decreased to 1.2 square inches and the actuator moment arm was decreased to 5.7 inches.

The moment of inertia of the gun was simulated by installing an eight foot length of four inch diameter barstock in the gun cradle. The test setup is shown in Figure 32.

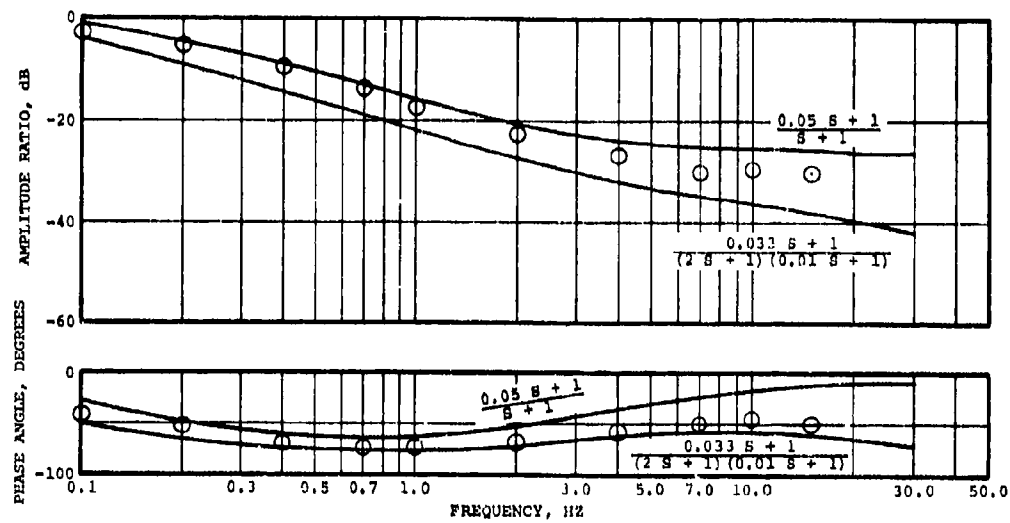


FIGURE 28  
FLUIDIC LAG-LEAD RESPONSE FOR  
GUN STABILIZATION SYSTEM

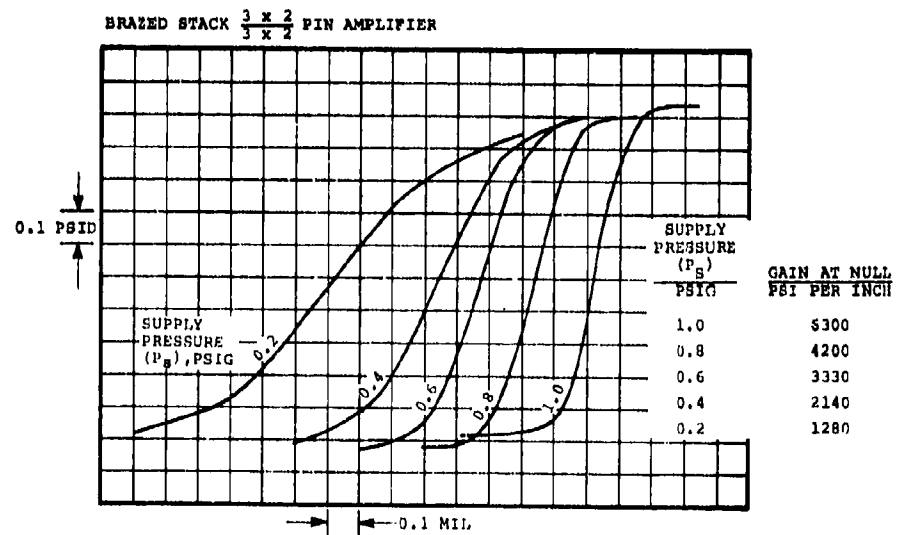


FIGURE 29  
STATIC PERFORMANCE OF  
CONTROLLER CIRCUIT

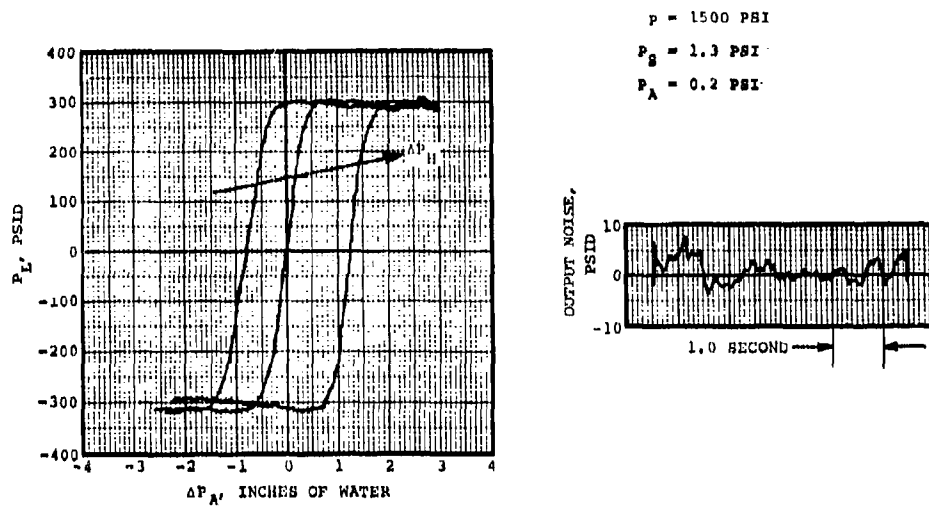


FIGURE 30

STATIC PERFORMANCE OF  
 CONTROLLER SERVOVALVE  
 (HYDRAULIC OUTPUT PRESSURE VERSUS  
 PIN AMPLIFIER OUTPUT PRESSURE)

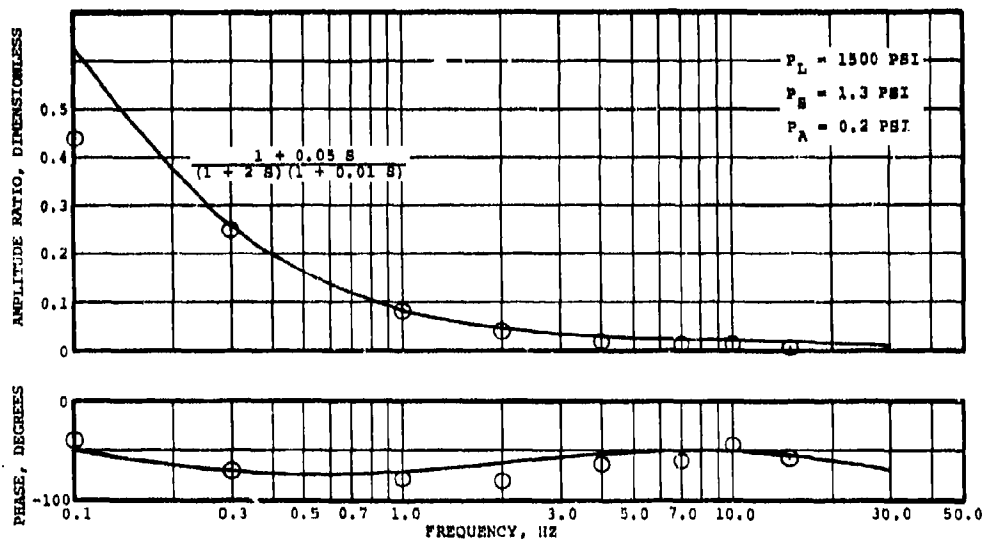


FIGURE 31

DYNAMIC PERFORMANCE OF  
 CONTROLLER/SERVOVALVE  
 (HYDRAULIC OUTPUT PRESSURE VERSUS  
 ELECTROFLUIDIC TRANSDUCER INPUT CURRENT)

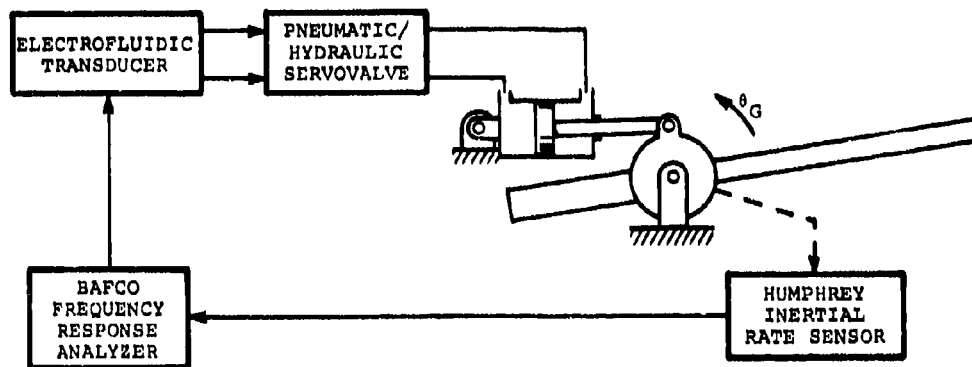


FIGURE 32

#### TEST SETUP FOR OPEN LOOP FREQUENCY RESPONSE TEST

The angular rate of the gun at various frequencies was compared to the electrofluidic transducer command signal by means of the Bafco frequency response analyzer. The results of this test are plotted in Figure 33.

As evident from the data, the time constant,  $\tau_G$ , of the actuation system was 0.16 second rather than the expected value of 0.7 second and was accompanied by an apparent transport delay of 0.02 second. This discrepancy was attributed to the presence of coulomb friction in the system, the value of which was determined by measurement to be approximately 46 foot-pounds. If the rate amplitude is assumed to be 0.25 radian per second, the equivalent friction coefficient,  $b_y$ , is determined to be approximately 68 pound-second per inch instead of the value of 0.15 pound-second per inch which was assumed in the analysis.

#### 4.3 Closed Loop Testing

It was analytically determined that, because of the discrepancy in the value of the time constant,  $\tau_G$ , of the actuation system, performance of the controller as initially designed would be poor.

Because of the lack of time, it was not possible to make a major readjustment of the dynamic compensation circuit to accommodate the lower value of  $\tau_G$ . However, analysis showed that disturbance suppression could be enhanced somewhat at low frequencies (0.1 to 1.0 Hz) by increasing the accelerometer damping ratio. This is shown in Figure 34. Since this adjustment was easily implemented, the system was tested under these conditions.

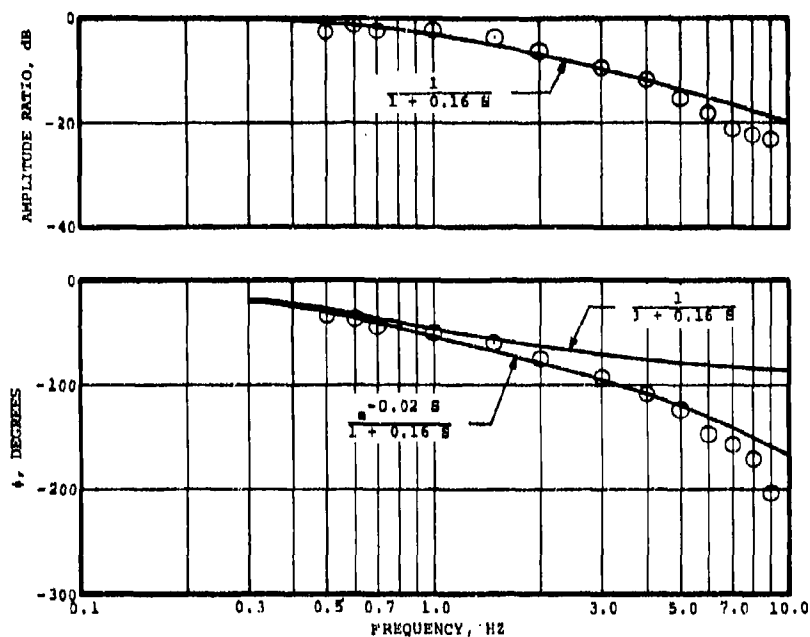


FIGURE 33

GUN RESPONSE TO SERVOVALVE INPUT  
(4 INCH DIAMETER X 7 FEET LONG  
BARSTOCK IN GUN CRADLE)

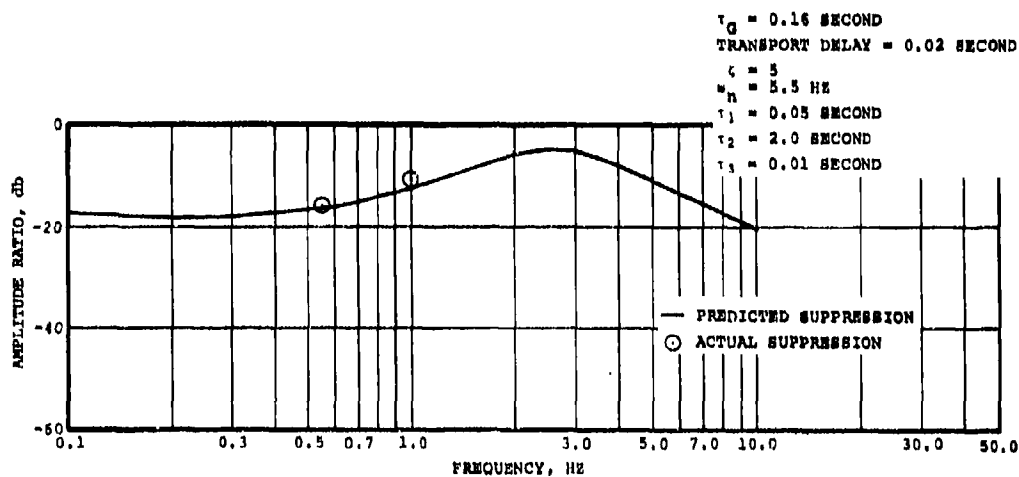


FIGURE 34

CLOSED LOOP  
DISTURBANCE SUPPRESSION WITH  
GUN STABILIZATION SYSTEM

The resulting disturbance suppression performance is plotted in Figure 34. Photographs of the turret test fixture and the controller installation are presented in Figures 35 and 36, respectively.

## 5. CONCLUSIONS AND RECOMMENDATIONS

### 5.1 Conclusions

The following was accomplished during the program:

- An analytical method was developed that can adequately predict the performance of a gun stabilization system. The linear frequency response model is of value because of its ability to quickly determine the effects of parameter changes on system performance. The more detailed analog computer model permitted a more precise evaluation of system performance because of its ability to include threshold and saturation non-linearities. The presentation of the analytical results in frequency response form rather than in the form of a step response or a random bump course permits objective comparison of the performance of various system concepts based on overall disturbance suppression performance. Both the linear and analog models can be extended to include more complex plant dynamics such as bending of the gun barrel.
- An accelerometer-controller unit was designed and fabricated to conform to a predetermined control equation. Advances were made and experience gained in working with pneumatic fluidic elements in the laminar flow regime. Use of a pin amplifier as a position sensing device for the accelerometer proved to be satisfactory. The commercial spring flexure used in the system was suitable for demonstrating system feasibility. However, as anticipated, this device will not be suitable for use in a production system because of the low natural frequency exhibited in a cross-axis direction. Also as anticipated, the magnetic fluid damping mechanism exhibited fairly severe temperature dependence.

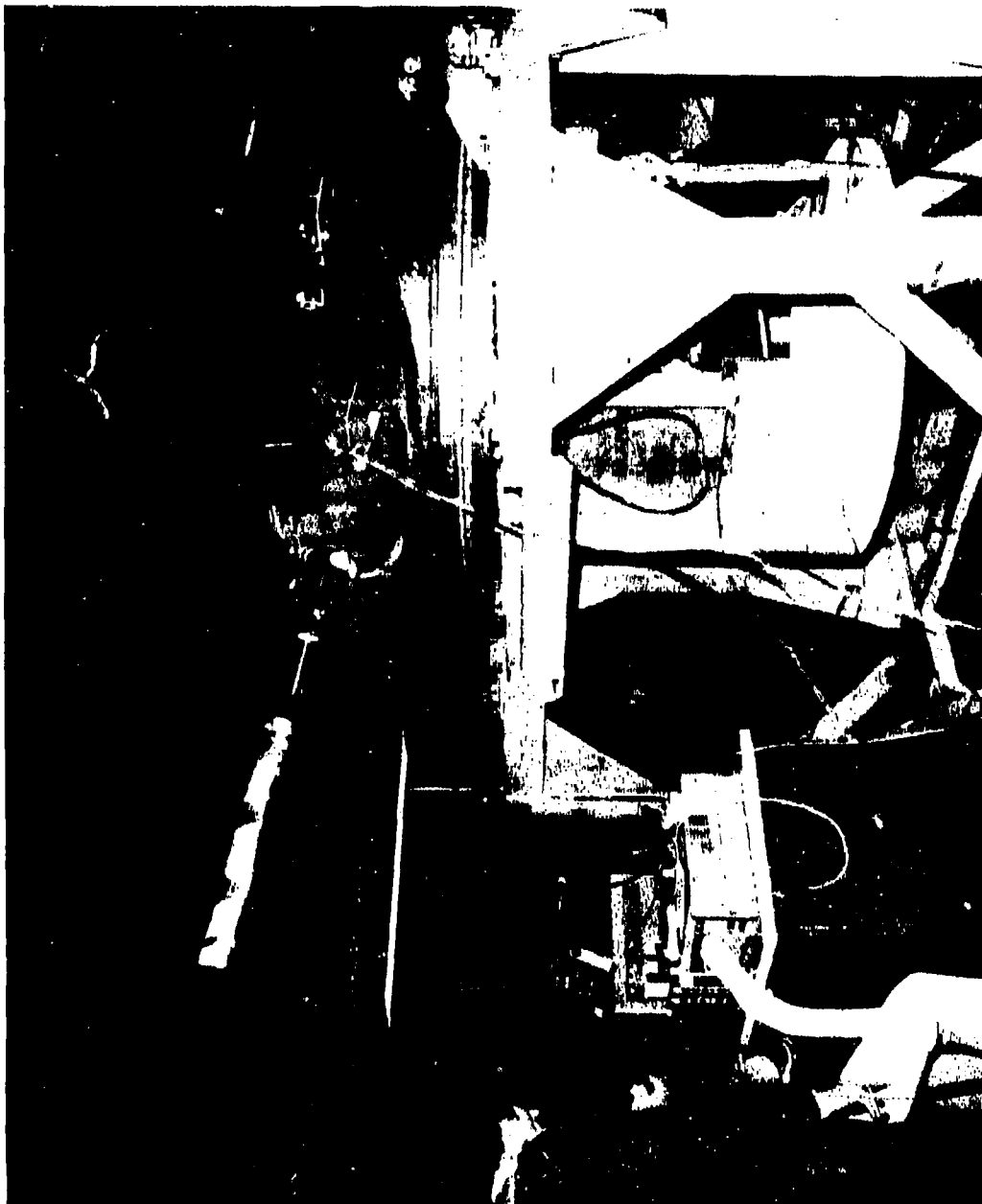


FIGURE 35  
TURRET TEST FIXTURE  
HDL FLUIDIC GUN STABILIZATION SYSTEM PROGRAM





FIGURE 36  
TURRET TRUNNION AREA  
SHOWING CONTROLLER INSTALLATION  
HDL FLUIDIC GUN STABILIZATION SYSTEM PROGRAM

- The desired disturbance suppression ratio of 20 decibels was not attained during the program. This was attributed to the assumption of a plant dynamic model which was inaccurate, resulting in improper design of the dynamic characteristics of the controller. Coulomb friction, of a significant and unanticipated magnitude, was present in the system. However, after the plant dynamics were accurately determined, the analytical model accurately predicted the actual performance of the system.

## 5.2 Recommendations

On the basis of the foregoing, it is recommended that, before designing a controller for a particular vehicle, an accurate determination be made of the dynamic characteristics of both the gun and the turret actuation system as well as the characteristics of the vehicle suspension system. It is also recommended that closed loop testing be included early in the development program to uncover and correct any unanticipated control system anomalies.

APPENDIX A

LINEARIZATION AND SIMPLIFICATION  
OF A TANK GUN ACTUATION SYSTEM

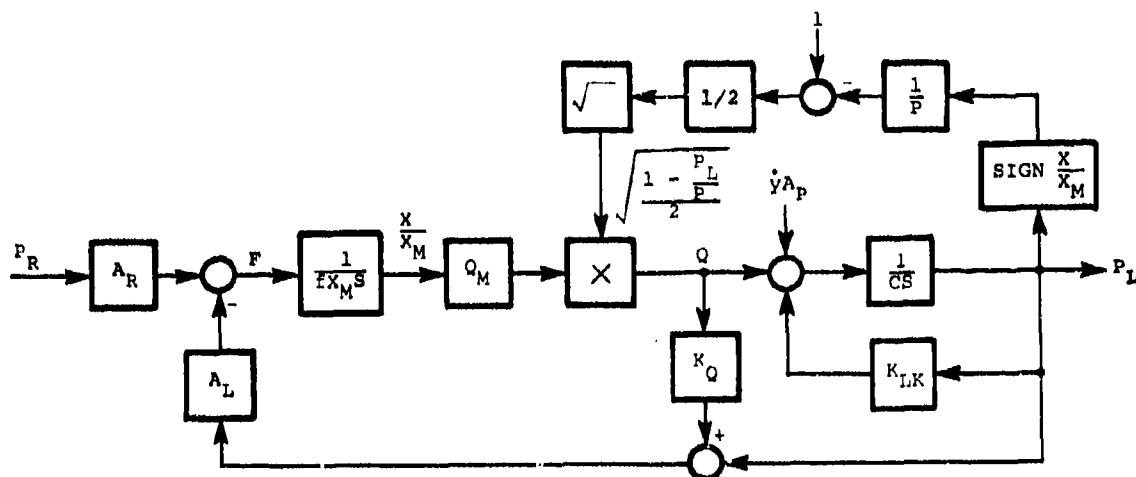
APPENDIX A

LINEARIZATION AND SIMPLIFICATION  
OF A TANK GUN ACTUATION SYSTEM

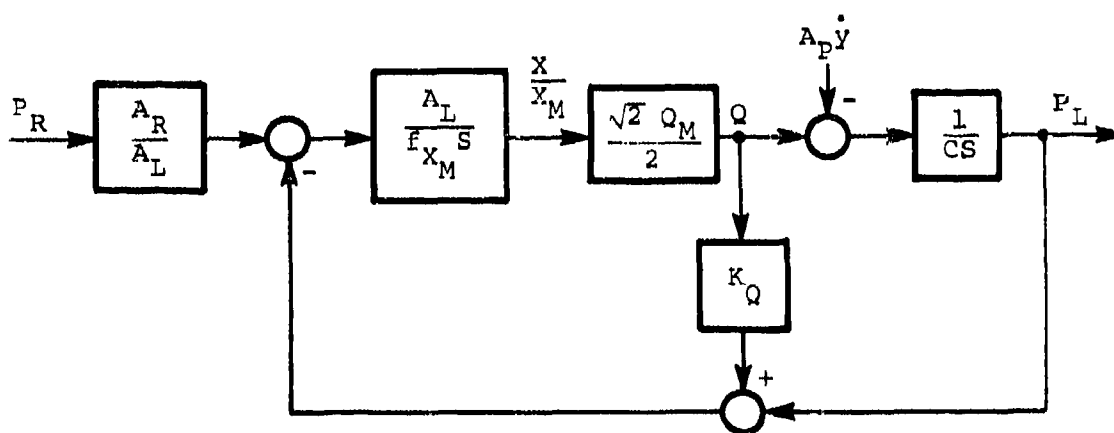
The pressure control servovalve is characterized by the following:

- $P_R$  command pressure differential, psid
- $P$  supply pressure = 1500 psi
- $P_L$  load differential pressure, psid
- $Q$  valve flow rate, cubic inches per second
- $K_Q$  droop constant, 20 psi per cubic inch per second
- $A_L$  area of pressure feedback land = 0.00985 square inch
- $A_R$  area of pressure command land = 0.0432 square inch
- $A_p$  piston area = 2.0 square inches
- $\dot{y}$  piston velocity, inches per second
- $F$  force on spool, pounds
- $f$  friction constant on spool,  $\frac{\text{lb-sec}}{\text{in.}}$
- $x$  spool displacement, inches
- $x_m$  maximum spool displacement, inches
- $Q_m$  maximum flow rate = 38.9 cubic inches per second
- $C$  hydraulic capacitance, cubic inches per pound

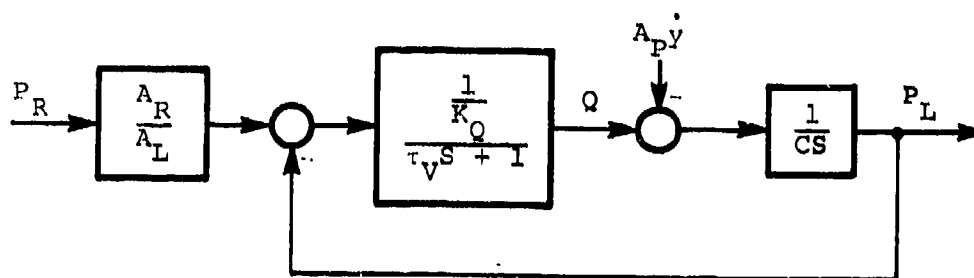
A block diagram of the system is presented in the following sketch:



If it is assumed that only small perturbations of  $P_L$  about  $P_L = 0$  are to be considered, and if leakage is neglected, then the block diagram can be reduced to:



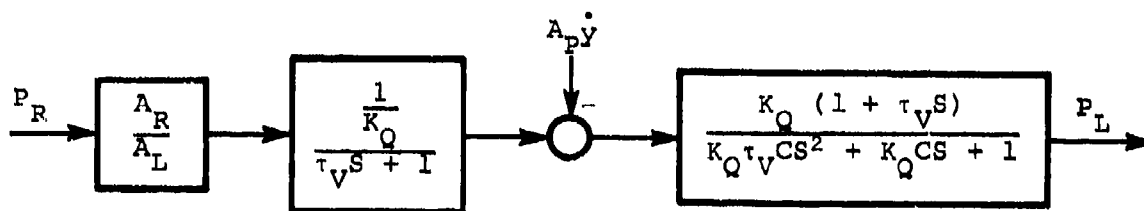
This further reduces to:



$$\text{where } \tau_V = \frac{f X_m}{Q_m A_L K_Q \sqrt{2}} .$$

The value given by MOOG for this time constant is  $\tau_V = 0.016$  second.

Further reduction of the block diagram yields the following:



The capacitance associated with actuator volume in the MIC-V tank is given as,

$$C = 3.23 \times 10^{-5} \frac{\text{in.}^5}{\text{lb}}$$

Therefore,

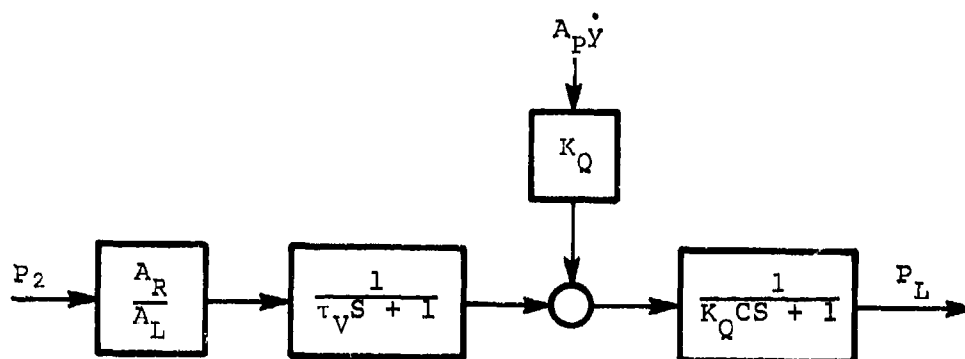
$$\omega_n = \sqrt{\frac{1}{K_Q \tau_V C}} = 1080 \text{ radians per second;}$$

$$\zeta = 0.349 = \frac{1}{2} \sqrt{\frac{K_Q C}{\tau_V}}$$

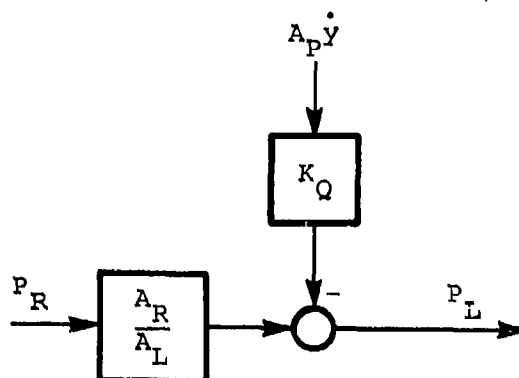
If the approximation is made that

$$K_Q \tau_V CS^2 + K_Q CS + 1 = (\tau_V S + 1)(K_Q CS + 1),$$

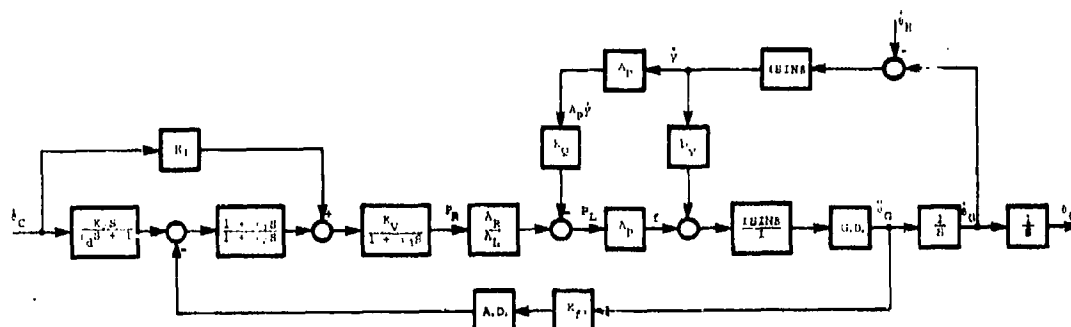
the block diagram reduces to:



If further simplification is desired, the small time constants  $\tau_V$  and  $K_Q C$  may be ignored:



This simplified valve model is included in the following block diagram of the gun stabilization system.



G.D. gun dynamics (bending)

A.D. accelerometer dynamics

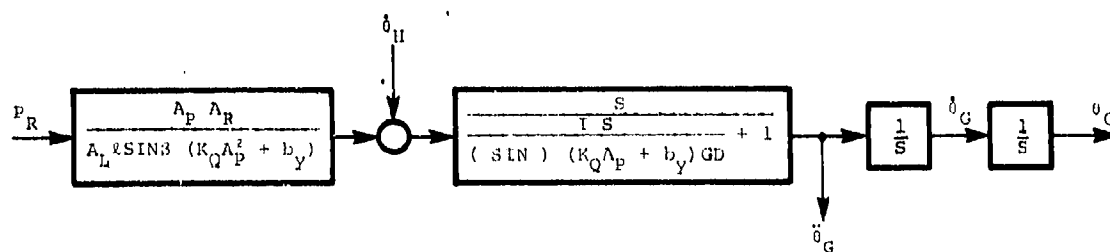
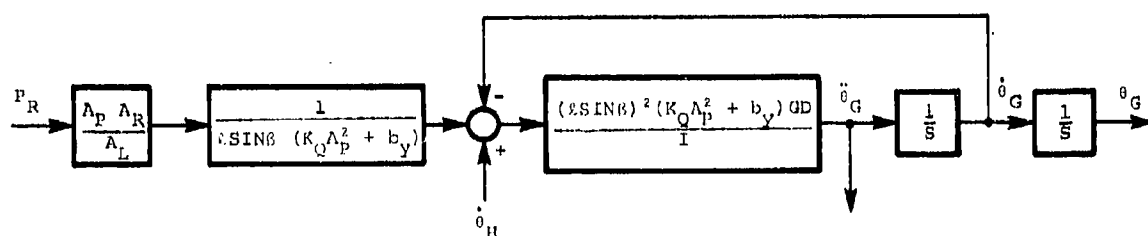
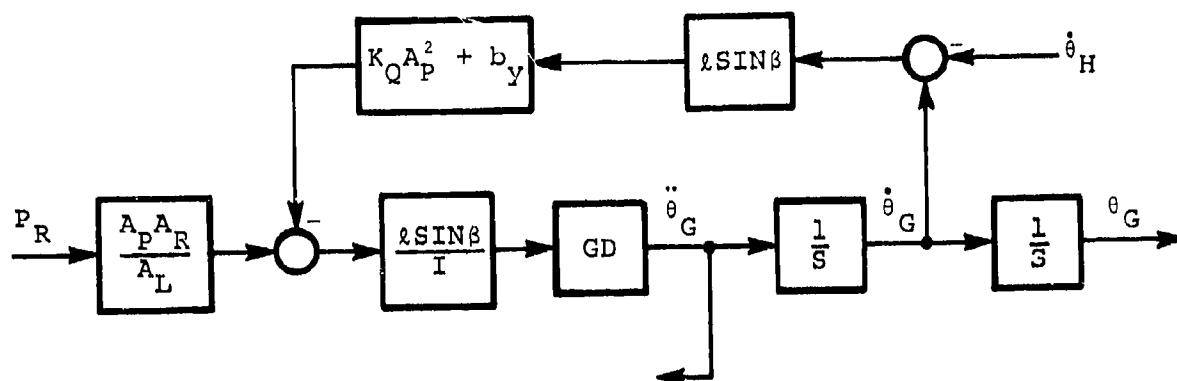
$b_y$  actuator friction coefficient

$\beta$  angle between actuator centerline and line connecting trunnion and actuator/gun attachment point

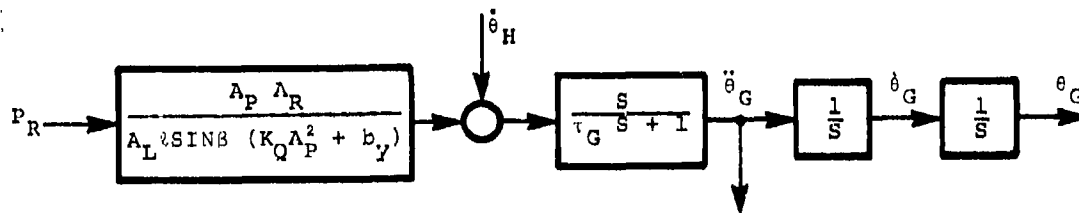
$z$  distance between trunnion and actuator/gun attachment point



The servovalve/actuator/gun loop may be simplified as follows:



Ignoring gun bending: (GD = 1)



$$\text{where } \tau_G = \frac{I}{(2 \sin \beta)^2 (K_Q A_P^2 + b_Y)}$$

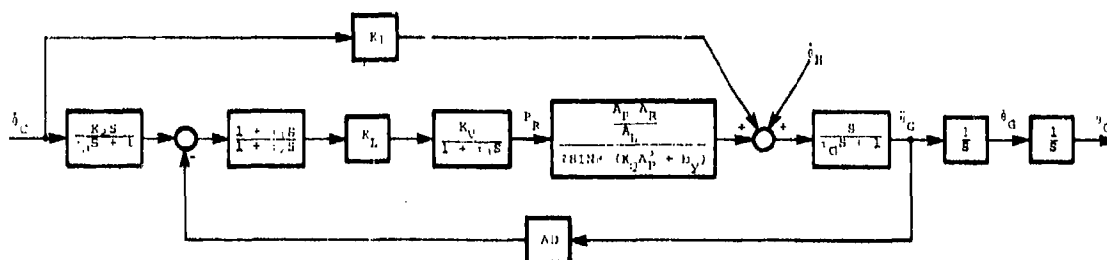
For MIC-V:

$$\tau_G = \frac{39.7 (12)}{[8(0.82)]^2 [20(2)^2 + 0.15]} = 0.13 \text{ second}$$

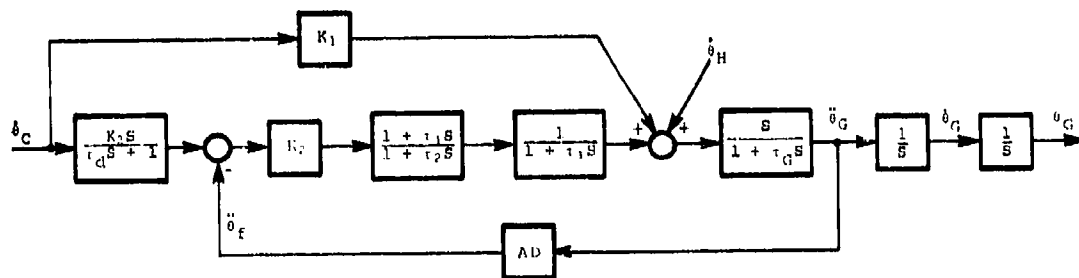
For M60A1:

$$\tau_G = \frac{3813}{(12 \cdot 3.2(0.4))^2 [11.11(4.72)^2 + 0]} = 0.78 \text{ second}$$

The gun stabilization system diagram with simplified pressure control servovalve, actuator and gun dynamics, and handle valve are shown below.



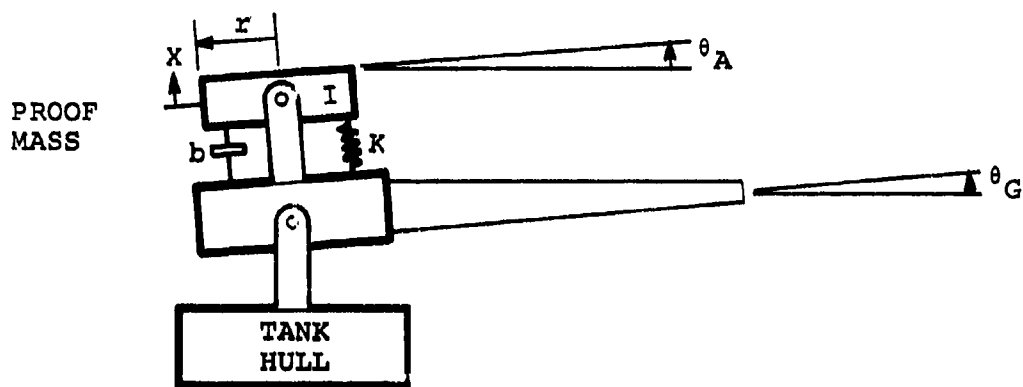
Defining  $K_v = \frac{A_L \lambda \sin \delta (K_O A_P^2 + b)}{A_P A_R}$



APPENDIX B  
ACCELEROMETER TRANSFER FUNCTION,  
GAIN, AND OPERATING RANGE

# APPENDIX B ACCELEROMETER TRANSFER FUNCTION, GAIN, AND OPERATING RANGE

Consider a proof mass accelerometer mounted on a gun, such that it senses angular motions of the gun about the elevation axis as shown in the following sketch:



where  $\theta_A$  and  $\theta_G$  are the angles of proof mass and gun centerlines with respect to the horizon,  $I$  is the angular moment of inertia of the proof mass,  $K$  is the angular spring constant, and  $b$  is the angular friction constant.

Motion of the proof mass is governed by the equation:

$$I\ddot{\theta}_A = -K(\theta_A - \theta_G) - b(\dot{\theta}_A - \dot{\theta}_G)$$

The following transfer function is then obtained:

$$\begin{aligned}\frac{\theta_A - \theta_G}{\ddot{\theta}_G} &= - \frac{1}{s^2 + \frac{b}{I}s + \frac{K}{I}} \\ &= - \frac{1/\omega_n^2}{\frac{s^2}{\omega_n^2} + \frac{2\zeta}{\omega_n}s + 1}\end{aligned}$$

where

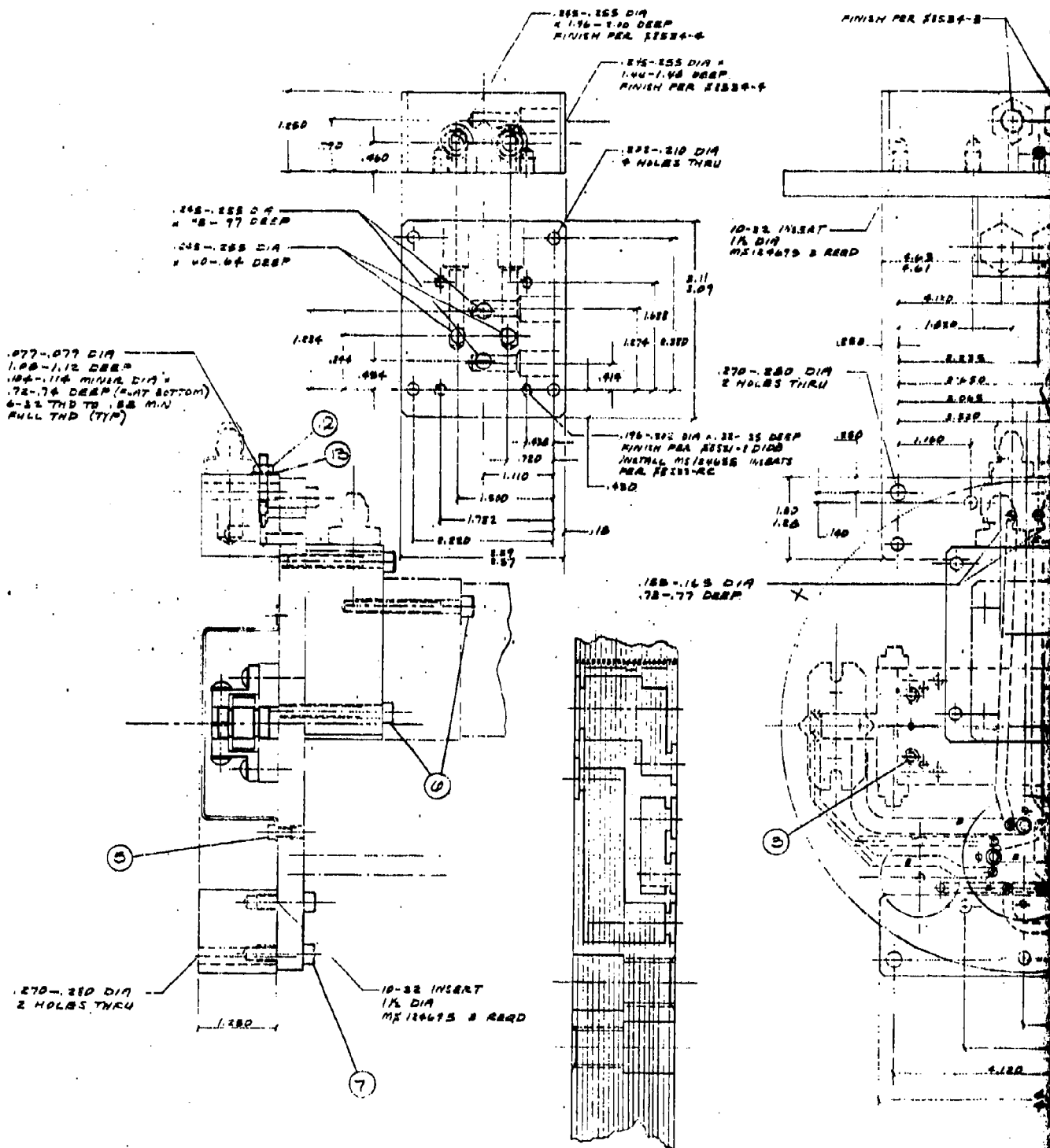
$$\omega_n^2 = \frac{K}{I}, \text{ and } \zeta = \frac{b}{2\sqrt{KI}}.$$

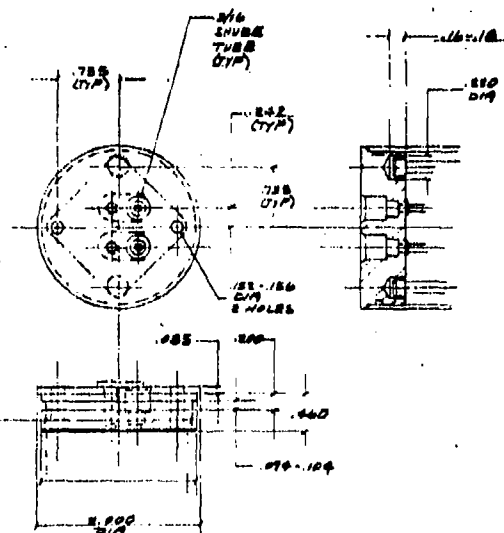
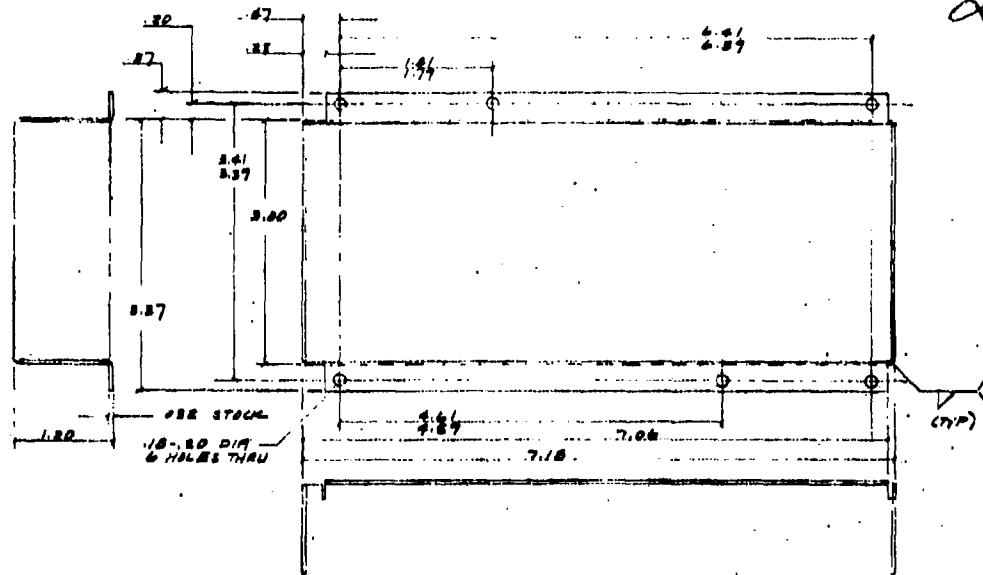
The relative angle,  $\theta_A - \theta_G$ , is determined by measuring the relative displacement  $X$  at the end of the proof mass so that

$$\frac{X}{\ddot{\theta}_G} = \frac{r}{\omega_n^2} \frac{1}{\frac{s^2}{\omega_n^2} + \frac{2\zeta}{\omega_n}s + 1}.$$

The steady-state sensitivity is:

$$\left. \frac{X}{\ddot{\theta}_G} \right|_{s \rightarrow 0} = \frac{r}{\omega_n^2}.$$



[illegible]



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